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THE DIESEL ENGINE SITUATION TODAY.

By MAX ROTTER, MEMBER.

In 1898 the first Diesel engine in the world, to be put into commercial service, was built by Adolphus Busch and installed in the Anheuser-Busch brewery in St. Louis. Between that time and 1912, the Busch organization designed and sold the only Diesel engines built in this country. In 1912, when the Busch-Sulzer company started its new factory, this type of engine was being built, both here and in Europe, solely for stationary service, and when the plant was laid out, the engineers considered themselves quite optimistic when they set as the probable limit in engine size, a cylinder of 400 B.H.P. and an engine of 6 such cylinders. Fortunately the overall dimensions and weight of this engine were determined on the basis of the then current piston speeds and heavy designs, so that the factory, as actually built, can turn out

engines of more than three times the then estimated probable limit of power. These circumstances are mentioned to bring out the phenomenal expansion that has taken place in the intervening 18 years.

Today there is not one field of power generation and use that has not been penetrated by the Diesel—from the aeronautic engine to the large stationary engine of 15,000 B.H.P. And by far the greatest and most rapid expension has been in the application of the Diesel to marine propulsion,—executed for the first time, in 1905, by Sulzer Brothers, in Switzerland, as a two-cycle; and in 1908 by Nobel, in Russia, as a four-cycle.

It is but fitting that this survey of the past include the recognition that is due, but has never been accorded to the American engineer who, at the very beginning, impressed his ideas upon the Diesel of two decades; namely James D. McPherson. It was he who designed the first trunk-piston type and the first multicylinder, vertical Diesels in the world; who developed a fuel pump and its governor control that are typical of those still being built; who showed, in his design of the combustion chamber, that he had a true understanding of penetration, diffusion, and turbulence that by many years antedated the adoption of these terms; and who installed the first Diesel engines in the world to run on coal tar, which they did successfully, for many years, in the plant of the United Gas Improvement Co., in Philadelphia.

McPherson's engines were the original type A, Busch-Sulzer. They were not as refined as modern designs; but 65 of them have been in regular operation more than 20 years; the dean being a 225 B.H.P. engine, installed in Texas in June, 1904, and still going strong. It is worth noting that the fuel consumption of these early engines was about as good as that of quite modern Diesels, namely .42 to .43 pound per B.H.P. hour. Incidentally, such records should serve to condemn the habit of many engineers, of estimating the useful life of a Diesel engine as 10 to 15 years.

It is a lamentable fact that, although it was this country that, in the beginning, established the type for Diesel engines, Europe has now far outrun us. Those who read trade journals that contain news of Diesel engines, cannot fail to be struck by the fact that we have not built a single stationary engine of over 4000 B.H.P.;

while, for example, Germany last year built two of 11,500 B.H.P. each, and is building 5 stationary engines of 5000 B.H.P. each for a single South American installation. In the marine field the comparison is still less pride-inspiring. Last year European countries completed 135 motorships of 3000 gross tons and over, with an aggregate propelling power of nearly 600,000 S.H.P. of Diesels, only 3 of which ships were for American account; while we did not turn out a single such motorship, our output being limited to the conversion, by the Shipping Board, of 10 ships, of a little over 6000 gross tons each, with a total of less than 40,000 S.H.P. Among the European-built ships are notable the *St. Louis* and *Milwaukee,* each of 17,000 gross tons, and each powered by 12,500 B.H.P. in M.A.N. 2-cycle Diesels; also 3 ships for the New Zealand Shipping Company, each of 17,600 gross tons, and each powered by two 5000 B.H.P., 5 cylinder, 2-cycle, Brown-Sulzer Diesels. And none of the foregoing figures include Japan, which last year completed the *Asama Maru,* of 16,500 gross tons and 15,000 S.H.P., in 2-cycle Sulzer Diesels.

It would be ridiculous to assume from the foregoing that we, in this country, are incapable of producing large Diesels, equal to those built in Europe. We have always excelled in large things. In the sister industry we have designed and built the world's largest gas engines, which are operating successfully on leaner gases than used anywhere in Europe. That the American Diesel industry generally turns to Europe for its large engine types, and restricts itself to adapting these to American methods and conditions, is natural, in view of the limited demand we have enjoyed, which has not justified the expense of independent development. Even our Navy has adopted a German design of submarine engine. One wonders whether our shipping industry is following the same fatuous course it did 70 years ago, when it lost its supremacy on the seas by clinging to the sailing clipper, of which it was justly proud, while Europe adopted the steamship.

While we are marking time in large Diesels, we have, in the past few years, proceeded very rapidly in engines of smaller capacities. The Diesel locomotive is making headway in capacities up to 1000 B.H.P.; the Diesel railcar is an established vehicle, the aeronautic Diesel has been perfected, and the automotive Diesel is coming to the fore. And it is not egotistical to assert that we

are doing more research work, relating to liquid fuel injection and combustion than any other country.

While the investigation of the processes of fuel injection is far from complete, and has unfortunately been confined to airless injection, we are learning the limitations of jet penetration and diffusion,—knowledge that is vital in the design of large airless-injection engines, and explains why such large engines generally show mean pressures much below those attained in their smaller prototypes. At present, airless injection engines with cylinder bores larger than 24 inches show mean indicated pressures below those attained with air injection; while in smaller engines the reverse is the case. These researches also explain the proved desirability of imposed turbulence in an airless injection engine; and give us a clearer understanding of some of the functions performed by the spray air of an air-injection engine, beyond that of merely spraying the fuel into the cylinder; namely,—the creation of turbulence and the improvement of diffusion by the intimate mixing of air and fuel during injection, and the expansion of the spray air, as well as its jet velocity as it enters the cylinder.

Differences of opinion have developed concerning atomization and combustion, and, while some of them may be academic, they are interesting. As a matter of fact many will probably affect the design of Diesels as little as the theories of water drop formation in the vapor affect the design of condensers. Little concerning atomization has yet been established beyond the fact that the drops of fuel break up as they travel through the compressed air. Undoubtedly a certain amount of atomizing is caused by the sudden expansion of the fuel, as it leaves the nozzle. After that, one theory is that the pressure against the leading side of the drop flattens it, until the fluid tension is overcome, and the drop breaks up; another, that the velocity of the drop creates a pressure reduction behind it, the drop assumes a streamline shape, and its trailing end is gradually triturated. Then there is the question whether vaporization or gasification precedes combustion, or the drop burns directly on its surface. Intricate calculations of flame propagation, which were inherited from gas engine theories, seem to have been wasted effort. It may even be doubted that combustion proceeds according to the commonly accepted picture; that is, accompanied by

projected flames, possibly it proceeds in the manner in which finely divided iron combines with oxygen, when introduced into it. That it is flame that issues from a cock in the cylinder may be due to the lower outside pressure.

The remarkable thing is that these processes can be accomplished in the time available for them,—sometimes little over two thousandths of a second. Without practical demonstration this would seem to indicate the advisability of preventing the atomized fuel from striking a surface by which its combustion would be retarded until it is decomposed; nevertheless there is evidence that the drop may strike a hot piston surface and be broken up, the smaller drops rebounding from it,—due either to their resilience, or to the pressure of vapor formed under and within them, as a drop of water explodes on a hot stove lid. Likewise, the drops of fuel should not collide, unless while their velocity is still high enough to disrupt them, and before it is low enough to allow them to coalesce.

In conjunction with these questions of injection, it is interesting to consider that of excess air; that is, the large percentage of the air, contained in a cylinder, that is not required for the combustion of the fuel. This excess air is generally about one-half of the total,—sometimes nearly two-thirds, and rarely less than 40 per cent. The natural snap judgment is to ascribe this to lack of perfection in the fuel-injection process, and to assume that perfecting the process would afford some of the advantages now attained by supercharging. While it may be true that, even if it were desirable, we are not yet able to consume a greater proportion of the air and attain smokeless combustion; it is doubtful that we could, at this date, go much farther in that direction, being limited by the temperature resistance of available materials; and it is equally true that we could not attain anywhere near the complete utilization of the oxygen, without a substantial loss in overall efficiency, and without an increased engine cost. Ordinarily the air, in a Diesel cylinder, is compressed to about 6½ per cent of its original volume, attaining a pressure of 480 pounds absolute, and a temperature of 880 degrees F.;—in this air is burnt a given weight of fuel,—say at normal engine load about one-half of the weight for which the air could supply the necessary oxygen,—the volume of air and gases expands to about 2½ times the

original volume of compressed air, with little increase in pressure; but with an increase in temperature to about 2700 degrees F.; the air and gases then expand about 6⅓ times in volume, and to a pressure of about 50 pounds absolute, performing work equivalent to a mean indicated pressure of about 115 pounds per square inch of piston area. Should 50 per cent more fuel be burnt in the same quantity of air, the temperature at completion of combustion would increase to at least 3700 degrees F.,—too high for ordinary present day materials and design. These conditions do not exist in a properly proportioned supercharged engine, in which the air excess is considerably greater than in a non-supercharged.

Supercharging,—by which is understood the artificial increasing of the weight of combustion air with which the cylinder is charged before the fuel is injected,—is gaining some adherents. Witness the propelling engines installed by Harland & Wolf in the motorship *Llangibby Castle,* each engine normally rated at 3200 B.H.P., and capable of developing 4250 B.H.P. when supercharged. The general idea of supercharging is quite old;—twenty years ago Sulzers were experimenting with it, but were turned aside temporarily by more urgent lines of development,—such as the 2-cycle engine.

There are several systems of supercharging. The best known are the Rateau and the Buechi, both of which provide the supercharging air by means of centrifugal blowers, driven by turbines using the exhaust gases of the engine as motive fluid. In general, these systems are similar to that developed in this country, by the G. E. Co., under Dr. Moss, for supercharging the gasoline engines of air craft, to maintain sea level air conditions in the engines, at high altitudes, and thus enable the engines to carry almost full load regardless of the rarified atmosphere. As stated before, by proper adjustment of the compression ratio and the air excess, these supercharged engines are not subjected to higher temperatures or higher maximum pressures than those existing in similar normally charged engines; although, of course, the average cylinder pressures are higher. The pressure of the supercharging air is about equal to the pressure of the exhaust gases—generally 18 to 20 pounds absolute; and the capacity increase of the engine, 30 to 40 per cent.

In the Buechi system the speed of the blower and therefore the pressure of the supercharging air automatically adjust themselves to the existing engine load, due to the variation of the exhaust pressure with the load. The result is high efficiency, and a very flat fuel consumption curve. This system is, therefore, suitable for continuous operation. To date it has been applied exclusively to 4-cycle engines; although it might well be applied to certain types of 2-cycle. The reason why the latter has not been done lies in conflicting patent interests.

Distinguished from the former systems of supercharging, intended as a regular condition of operation, is the Sulzer system, intended primarily to increase the overload capacity of a 2-cycle Diesel Engine. In this system the supercharging air is furnished by a compressor, attached to the engine, which compressor comes into and goes out of action under control of the engine governor; so that supercharging takes place only at the higher engine loads. The automatic valves in the upper tier of the Sulzer scavenging ports, for the normal scavenging and charging, are supplemented by timed valves, controlling the delivery of supercharging air to the cylinder, so that the latter air, furnished by the supercharging compressor, is admitted only towards the end of the charging period; that is, after the piston has covered the exhaust ports, and before it covers the upper scavenging ports. With this arrangement the capacity and power consumption of the supercharging compressor are reduced to the minimum. The capacity of the engine may temporarily be increased as much as with the Buechi system; but the fuel efficiency of the engine is adversely affected to the extent of about 5 per cent, due to the power consumed by the supercharging pump, even when not in action. However, it is an inexpensive means to increase the peak capacity of an engine, and when the available space does not permit the installation of a larger engine.

Still another system of supercharging has been applied to certain German 4-cycle engines, for the same purposes as accomplished by the Sulzer system. In this system also, the supercharging air is furnished by a compressor attached to the engine. In addition to the normal admission valve, a timed supplementary piston valve is provided, which gives access of atmospheric air to the admission valve during the earlier portion of the suction stroke,

and of supercharging air during the later portion of the stroke. This, as in the Sulzer system, reduces the dimensions and power consumption of the charging compressor, below what they would be if all of the charging air were furnished by it; but it, too, increases the fuel consumption at all loads.

As in the Buechi system, the compression and air excess in the Sulzer and German supercharged engines are adjusted so that the combustion temperatures will be substantially the same as in the un-supercharged engine.

The process of scavenging,—that is, clearing the cylinder of the products of the previous combustion,—has a close relationship to supercharging, and is receiving considerable attention.

The scavenging of a 4-cycle engine is an obvious process;—when the exhaust valve is opened, the major portion of the products of combustion are expelled from the cylinder by their expansion; the remainder is pushed out by the piston. This necessitates opening the exhaust valve against the terminal pressure in the cylinder,—about 35 pounds,—requiring rugged valve gearing. Attempts have been made to reduce this work by providing exhaust ports, uncovered by the piston near its outer dead center; but the complication of this construction has not been justified by its advantages. A recent invention provides a smaller relief valve, within and opening in advance of the main exhaust valve; but it is almost a foregone conclusion that this, also, will not be worth while.

The scavenging of a 2-cycle engine is a more difficult problem, including, as it does, the simultaneous recharging of the cylinder. These processes must both be accomplished in 40 to 50 degrees of crank travel, as compared with 360 degrees available for the same, in a 4-cycle engine. Moreover, the process is not a definite mechanical one, as it uses air to displace the spent gases, and it has therefore been and still is the subject of considerable experimentation, including, as it does, the creation of the desired character of turbulence. All such experiments suffer from the same limitations,—they are made with a substantially constant stream of air, through a constant cylinder volume, against a constant pressure, through constant openings, without disturbing mechanical motions within the cylinder. They have been of value, however, in pointing the direction of "cut and try" experiments; but can

scarcely be considered as more than remote indications of processes that are complicated by the variation in pressures, and stream velocities and directions, caused by the progressive uncovering and covering of ports by the piston; by the variable cylinder volume; and by the already existing turbulence of the gases due to their expansion and outflow. Every catalogue picture, illustrating a scavenging system, shows clean-cut curves and loops, embellished with arrowheads,—sometimes in two colors, red for the hot exhaust gases, blue for the pure cold air,—which indicate what might reasonably be expected if the cylinder were a vertical slice through a hemisphere, with a scavenging port to the East, and an exhaust port to the West. It is a question what actually happens when a number of other streams are, all at the same time, trying to blow across the North pole from all points East of Greenwich, to all points West,—sometimes detouring around the tropics, while the tropics are oscillating between Capricorn and Cancer.

It does not appear that any Diesel designer has yet attempted to scavenge and charge a 2-cycle cylinder through exhaust and scavenging valves, both in the cylinder head. All other combinations include piston-controlled ports, either for exhaust, or for scavenging, or for both. Most of the earlier 2-cycle engines exhausted through ports and admitted scavenging air through valves in the head; but this arrangement has been almost universally abandoned. Reversing the process, we now find the new B. & W. two-cycle, double-acting engine admitting scavenging air through piston-controlled ports, and exhausting through timed valves in the cylinder heads. This,—including the use of the exhaust valves as supplementary power pistons,—is decidedly the most novel development of last year, and the performance of the engine in the *Amerika* will be watched with keen interest. Its reported efficiency and capacity are remarkable for a double-acting, airless injection engine.

The simplest arrangement has been most generally adopted,—piston controlled ports for the scavenging air around one-half cylinder circumference, and the same for the exhaust around the opposite half; but in at least one construction the scavenging and exhaust ports alternate around the whole circumference. Usually the exhaust ports are first uncovered by the piston, and the scavenging ports somewhat later, the interval being sufficient to allow

the pressure in the cylinder to fall below that of the scavenging air. This arrangement affords the least time for charging, and under certain conditions allows exhaust gases to blow back into the cylinder. An improved condition is obtained, in engines like the Fiat and Setz, by locating the edges of both scavenging and exhaust ports so that they will be uncovered simultaneously, and providing the former with inwardly opening valves. Neither of these arrangements enable the charging of the cylinder to a pressure higher than the exhaust back pressure,—in fact the eductor action of the exhaust may cause an under-pressure in the cylinder charge. There are examples of increasing the back-pressure, by throttling the exhaust; but this is liable to affect the scavenging adversely, and should be resorted to only to assist in equalizing the work among the several cylinders.

The arrangement that most nearly fulfills the desired conditions is that developed by Sulzers, and improved by Reale and Busch-Sulzer, in which the scavenging ports are arranged in two tiers, the upper being provided with automatic inwardly opening valves, and being uncovered before and re-covered after the exhaust ports, thus providing the maximum time for charging.

A marked tendency at this time is towards airless injection. Although this system has long been in use on engines of smaller capacities, and has been applied to large engines,—by M.A.N. occasionally, by Doxford more extensively, and with marked success in the A.E.G.-Hesselman engine,—its more general use in large engines has recently increased notably. A comparison of the performance of air injection and airless quickly dispels the former common belief that the efficiency of the airless is higher to the extent of the power consumed by the injection air compressor; which consumption has ordinarily been stated to be 10 per cent of the power developed by the engine. The average airless-injection engine actually consumed only about 5 per cent less fuel than an equivalent engine with air-injection.

The injection of 1 pound of fuel requires about 20 cubic feet of free air, compressed to 800 to 1000 pounds pressure. If these quantities are considered as being per minute, they would apply to an engine of about 150 B.H.P.; and the compression of the spray air would consume about 10 B.H.P. In ordinary practice, adjustments are not made so closely, and the compressor load may

readily be 50 per cent higher than this, apparently justifying the stated consumption of 10 per cent. The spray air is compressed in three stages, and cooled to about atmospheric temperature before it reaches the fuel valve. Thus the conditions are, roughly, that air at say 900 pounds pressure and atmospheric temperature in the fuel valve, enters the power cylinder with a pressure drop to say 475 pounds, there its temperature is raised by the compressed air in the cylinder, to about 840 degrees F. From this point the spray air acts as part of the compressed air charge in the cylinder, expanding and performing work on the piston, just as does the air compressed in the cylinder itself, whether the injection be by air or airless. The actual power chargeable to the compressor is, therefore, its friction (say 2½ horsepower) plus the power consumed in compressing the spray air from 475 to 900 pounds (say 1½ horsepower); this sum, liberally figured, is thus less than 3 per cent of the net engine output. The weight of the air in the compressor space of the engine is about 30 pounds per pound of fuel, and its temperature, as stated, about 880 degrees F. before being cooled by the injection air. The acceleration and atomization of the fuel consumes about the same power as in an airless-injection engine,—say 3/4 horsepower per pound of fuel per minute,—and the temperature of the whole cylinder contents is reduced about 40 degrees F. To overcome this and place the engine on a par with airless injection, the compression in the power cylinder of the air injection is increased about 20 pounds, adding to the work of compression about 2½ horsepower, which is absorbed by the cooling effect of the spray air. Thus, the total power consumption, in such an engine, chargeable to air injection, ranges from 6½ to 8 horsepower, or about 4½ to 5½ per cent of the engine capacity.

The foregoing accounts for the difference in fuel consumption, and the higher compression usual in air-injection engines. The figures are, of course, subject to modification according to the size of the engine; and the commercial value of the saving must be adjusted to the prices of the fuels that may be used successfully in the respective engine types. It must not be assumed that the higher efficiency, simplification, and reduced first cost of the airless injection engine are gained without some sacrifice in ability to use fuels that may be cheaper, and this disability may become

important if and when liquid fuels other than petroleum, shall be produced to sell at prices below those of petroleum;—and this is not an unlikely development.

Other tendencies in construction and arrangement, some of which may be overstepping the desirable limit, and about each of which much could be said if time would permit, are the double-acting engine; Diesel-electric propelling engines, as installed in the converted American ships *Triumph, Courageous,* and *Defiance,*—a type of propulsion that has not found favor in Europe; Diesel-geared propulsion, as installed in a number of large European ships, including the recently commissioned *St. Louis* and *Milwaukee,* with 12,000 S.H.P. each, and the much discussed *Ersatz Preussen,* with 50,000 S.H.P.; but in only one moderate sized American ship, the *Herman Falk,* with 2600 S.H.P. There is also the composite steam turbine and geared Diesel installation, in a naval cruiser, in which the center propeller, consuming 12,000 B.H.P., will be Diesel driven, to obtain high efficiency and long radius of action at the ordinary cruising speed of 18 knots and in which the two outboard propellers, each consuming 30,000 B.H.P., will be driven by steam turbines, to enable a ship speed of 32 knots when necessary.

It is natural that in an industry that has grown to the magnitude of Diesel manufacturing, there should be an urge towards standardization: and standardization confined to proper channels is most desirable; but it must not overflow into domains in which it would retard progress, or discountenance efforts towards improvement. For example, it is clearly most desirable to standardize the form and dimensions of articles, also the composition and properties of materials, if such articles and materials are extensively used; but it is most undesirable to standardize the application and conditions of use and service of such articles and materials. Where would the steam turbine industry be if the speed, steam pressure, and steam temperature conditions had been standardized or limited 25 years ago, leaving no incentive for engineering skill and metallurgical research, beyond that of endeavoring to reduce the cost of manufacture? Standardization must not be such as will abridge the effort to improve, as well as to reduce cost, by skillful design and the intelligent selection of materials. The recently suggested limitation or standardization of Diesel

engine ratings, on the basis of mean indicated pressures, would appear to come within this proscribed territory. This limitation has been predicated on the thesis that the Creator gave us certain materials, which possess definite properties;—it must not be overlooked that he gave us also brains to modify such properties by combinations and treatment of such materials, and to select materials according to their intended use and service. Consider, for instance, the piston head of a Diesel. It may be well designed or poorly designed; it may be cooled or uncooled; it may be made of ordinary or special alloyed cast iron; of forged or cast carbon or alloyed steel, or of aluminum or magnesium alloys; and superimposed on this choice of materials are infinite variations in molding, casting, and heat treating. In addition to which, every change in diameter and rotative speed presents changed problems. To require the application to all of these of a single rule, based on temperatures, is not reasonable.

A maximum basis of rating is not a protection for the purchaser. Its acceptance will not ensure good judgment or honesty,—any more than constitutional amendments and laws ensure abstemiousness and chastity. As in these, the violators will discount the probabilities of being caught, and there are conditions of service,—such as marine propulsion,—in which the actual power consumption cannot be readily established. A maximum rating is likely to be a standard rating, when the conscience is flexible; whereas the rating of an engine should express the work that particular engine may be depended upon to perform, with safety and reasonable wear, and with a fair margin to allow for possible dis-adjustments, with due regard for the available opportunities for correcting them. For this reason, Busch-Sulzer, for example, establish from 3 to 5 distinct ratings for one and the same engines,—the highest being for stand-by service, in which the engine may operate only a few days in the year; the lowest for single-screw ocean-going ship propulsion.

The variety of the bases of rating, proposed by British engineers, indicates the complexity of the matter. For example, they have suggested that the rated normal capacity of an engine be the power it will develop continuously for 12 hours (presumably with invisible exhaust), and that the continuous rating be 10 per cent below this; but they apparently recognize the deficiency of this

method, by limiting it to engines for driving electric generators. It has also been suggested that the ratings be fixed at established percentages below the stalling torque, or the load at which the engine definitely begins to drop its speed; but this torque and load will vary considerably with the immediate condition of the engine, the character of fuel and lubricating oils, and atmospheric conditions. And neither of these methods differentiate between different qualities of engines; nor does the third European suggestion that the rating be a definite percentage below the maximum load the engines will carry with smokeless exhaust.

The recent American proposal to limit Diesel ratings to certain maxima in mean indicated pressures,—assumed to reflect maximum permissible temperature stresses,—is unsatisfying for the previously stated reasons, and, furthermore, selects a criterion that is confessedly unreliable, excepting in slow-speed engines; but which it proposes to apply to engines of speeds up to 800 R.P.M. It would appear more logical to limit the amount of fuel burnt per minute per square inch of piston area; but this is not being advocated in this paper. It is rather the writer's belief that so long as the majority of purchasers compare engines on the basis of price per horsepower, per pound, and per cubic foot of cylinder volume,—which latter, by-the-way, a purchaser recently did without considering that he was comparing a 4-cycle with a 2-cycle,—it is each builder's job to justify his own rating, as applied to his engine for the contemplated conditions of service.

The foregoing leads us to another question:—What is the M.I.P. of a 4-cycle engine? This is important, in view of the practice of many consulting engineers of limiting this pressure and requiring a guarantee of brake mechanical efficiency. That such limitations and guarantees are of little value, without qualifications that would practically specify the make of engine, is beside the question. The purpose of the practice is obviously to limit the temperature stresses at the specified useful power output. Therefore, the method of measuring the M.I.P. should be such that this pressure will truly reflect the temperature conditions.

In a 4-cycle indicator card, the bottom loop area represents the work of pushing out the exhaust gases and drawing in the air charge. The more inadequate the valve areas and ports, the larger will be this work; and, if the area representing it be deducted

from the upper diagram area, to determine the M.I.P. and brake mechanical efficiency, the poorer the engine design the greater will be the area of the upper diagram for the same M.I.P. and efficiency, and the higher the temperatures. Moreover, the bottom loop represents the same work as is performed by the scavenging pump of a 2-cycle engine; yet it would be entertaining to hear the remarks of the purchaser's engineer if the builder of a 2-cycle engine were to deduct the mean pressure of the scavenging pump diagram from the mean pressure of the power cylinder diagram, in establishing his M.I.P. and brake mechanical efficiency, although there is little difference between the work consumed by the scavenging pump of a well designed 2-cycle engine, and that consumed in the exhausting and charging of a less well designed 4-cycle, and there is a substantial cooling effect exercised by the large volume of scavenging air passing through the 2-cycle cylinder.

DIESEL ENGINES IN SUBMARINES.

By E. NIBBS, MEMBER.

Since 1899 when the then Holland Torpedo Boat Company, using J. P. Holland's patents, first demonstrated that a vessel could be successfully and consistently controlled both on the surface and under the water, a never ending quest for improving the means of propulsive power was started.

I have a few slides which with your permission I will show at the end of this paper, but I must first state that the slides deal chiefly with mechanisms with which the Electric Boat Company (the successor of the Holland Torpedo Boat Company) is most familiar, although there are other mechanisms of possibly equal or greater merit, but with which we are not so familiar.

In Revolutionary times, a submarine was constructed, the propulsion equipment being one man power. He had a very good mechanical efficiency, but without much capacity for sustained effort. Later the vehicle expanded, and in the Civil War period, the power rose to eight man power, but it still possessed the original drawbacks. Then came the first really reliable submarine, and it was powered with a gasoline engine of two vertical cylinders of the four cycle type. It developed 45 horsepower and was built by the Otto Engine Company of Philadelphia. It was the prototype of several others with minor improvements, and with different numbers of cylinders, but these gas engines formed a strong example of arrested development, due to a realization of the fact that gasoline was a very objectionable fuel for use in a sub-surface vessel.

We can consider ourselves fortunate in that no major accidents resulted from the use of gas, but it was recognized that its employment constituted a grave source of potential danger. Hence submarine designers anxiously scanned the factories and research departments of the world, in order to find a heavy oil engine, which type was already emerging from the pangs of being born, though hardly out of swaddling clothes.

At this same period, the English submarines, employing the Holland patents, and which used at first the Otto engines, were using 12 and 16 cylinder petrol engines arranged horizontally on either side of a central fore and aft crankshaft. The B and C Classes of British boats were thus fitted. They worked well, back-fired joyously on starting, cracked cylinder heads, or breech ends as they were termed, diluted oil in the crankcase, occasionally had an explosion therein, and expended spark plugs at a fearsome rate. The carburetion was imperfect judged by standards even of those days, gassing by CO was relatively common, and eventually mice were carried to act as CO detectors, which duty they performed efficiently. Accidents when fuelling were too frequent and the structure of the tanks, which in some cases had water jackets surrounding them, was difficult to build and harder to keep tight.

The same difficulties that made us search for a Diesel engine caused the same search to be made in other countries and Vickers, in England, evolved the type of engine which was fitted in the British D Class, which supplanted the petrol engine entirely, and its development was followed in a single minded, direct fashion, so that the engine which started with air injection in the early D's changed to mechanical injection in the later E's and in all subsequent British built submarines.

In the beginning M.A.N., through the efforts of Drs. Bruns, Schwartz and Rimmel, had come to the conclusion that considerations of weight, space and power involved led to the adoption of the two cycle engine, as it apparently offered the greatest inducements, and engines operating on this cycle were designed, built and tested. Sulzer in Winterthur, Switzerland, and Krupp in Essen, were doing the same, as well as one or two other German firms. In 1913, it became evident to M.A.N. that metallurgical science and foundry practice had not advanced to the point reached by design, so Dr. Lauster of this company designed, built and tested at Augsburg in early 1914 a 1000 horsepower, four cycle engine, while still continuing to build the two cycle type for actual use in submarines up to the end of that period. This engine gave good results in shop tests and was installed in a submarine. In service, weakness developed in the housing, which was the conventional housing over the bedplate with cylinders set thereon,

and in consequence Dr. Lauster redesigned the housing arrangements, making the housing much deeper, used cylinder liners, and this engine in 1915 became the famous SV45/42, 1200 horsepower engine which was used in the majority of the German submarines during the World War. It was put into mass production and assembled by other firms than M.A.N. in order to obtain the numbers needed for military purposes, even as the Vickers mechanical injection engine was being built in England at the same time for the same purpose though on the opposing side.

The 45/42 was expanded into the 53/53 size, and also reduced to the 35/35. These engines were distinguishable from all others produced at that time, inasmuch as they had shop test of 72 hours full power (noting that a short enforced stop of limited time did not invalidate the trial). They carried a full load M.E.P. of 86 pounds per square inch and had the hitherto unprecedented piston speed of 1350 feet per minute, and were directly reversible, besides incorporating a variable lift of the spray valves, of which two were employed per cylinder. The engine construction consisted of light sectioned steel castings forming a rigid structure. Air injection was employed and the standard of workmanship used in the construction of these engines was of a high quality.

In these United States during the gasoline era, we looked for a heavy oil engine all over the world, and engines of the Vickers earliest type were built and fitted in the E and F boats. They did not at that time come fully up to the high expectations that had been formed, and so nothing further was done with their development on this side of the Atlantic. Finally, my company came to an agreement with the M.A.N. Co. to act as their licensees, which connection—interrupted by the war—exists today. We had the benefit of the M.A.N. experience up to 1914, which extended then only to the developed two cycle engines and these were built by us, and were fitted in the H, L and K Classes.

Busch-Sulzer, under the Sulzer-Freres license also built two and four cycle engines for submarines, and thus there was a healthy competition between two American firms, each specializing in a Diesel power plant for under-water vessels. It is noteworthy that both firms have built engines for submarines of the two and four cycle type, and it is a compliment to the open-mindedness of

the Navy Department that some vessels have engines of both two and four cycle type installed therein.

Schneider in France, Fiat and Tosi in Italy, and Burmeister & Wain in Denmark, had done a comparatively small amount of submarine work before the war, but the above describes the main pre-war activities.

During the War, Vickers in England made no essential changes, but concentrated on mass production, Sulzers in Winterthur developed the two cycle, single acting engines, chiefly as regards size and improvement in details. The Swedish, Norwegian and Danish firms developed commercial engines. Italy and France concentrated on what was already done and were chiefly concerned with output. The Germans concentrated on mass production of the M.A.N. four cycle trunk piston engine, necessarily stopping development of the Krupp design.

In the United States, the E. B. Co. developed a four cycle engine entirely independently of the M.A.N. Co., built an experimental single acting, two cycle, crosshead, air injection engine; a four cycle, double acting cylinder; a four cycle, six cylinder engine of which two were coupled to the same shaft; and finally, by force of circumstances, had to produce in quantity the majority of the main engines for the O, R and S Classes, which are of the four cycle, trunk piston type and which machinery is still giving excellent service in vessels afloat. Busch-Sulzer continued with two cycle development.

The next convenient period is that dealing with post-war activities. Since 1920, when Busch-Sulzer two cycle engines for the *V-1* to *V-3* were ordered, no private firm has had any orders from the U. S. Government placed with it. The Navy Yard at Brooklyn has built "Bureau Type" engines of the already developed M.A.N.-SV types in the various cylinder sizes. The E.B. Co. built and tested several single cylinders and reached a satisfactory conclusion with regard to a two cycle, double acting, moderately high speed type, and also with regard to supercharging four cycle type engines. I am unable to state what development has been done by Busch-Sulzer as regards submarine engines.

In Europe, it has been confined to—Germany, including associated firms in Switzerland; Great Britain; Italy; Switzerland; France.

The following firms are those responsible:

Germany—M.A.N.—Double acting, two cycle, crosshead, mechanical injection engines, and single acting, trunk piston, four cycle mechanical injection.

Great Britain—Vickers-Armstrongs Ltd.—Single acting, four cycle, trunk piston, mechanical injection engines.

Italy—Tosi—Single acting, four cycle, trunk piston and crosshead air injection engines.

Fiat—Single acting, two cycle trunk piston and crosshead air injection engines.

Switzerland—Sulzer—Single acting, two cycle, trunk piston and crosshead air injection engines.

France—Schneider—Single acting, trunk piston, two and four cycle, air injection engines.

As shown by the above list of designers and confirmed by examination of the practice of the different European navies, both two and four cycle, single acting engines are freely used in Europe; in fact, no single important European navy uses one cycle to the exclusion of the other, except the British Navy, which still adheres to the single acting, four cycle, mechanical fuel injection type. The British Admiralty has, however, developed by extensive experiments a double acting, two cycle construction for large powers. These experiments are described in the 1928 number of Institute of Naval Architects Proceedings.

To sum up, as to single acting engines, the choice between two and four cycle has not depended upon any inherent superiority of one cycle over the other, but on other considerations not entirely of an engineering character.

By reason of the small and moderate displacements and dimensions of the pre-war submarine, the power requirements were then best met by the use of single acting, trunk piston engines and accordingly the efforts of all designers and builders were then concentrated on such engines. The goal aimed at was the production of a compact high speed engine combining light weight with a satisfactory degree of reliability and durability. No light weight, high speed pre-war engine proved satisfactory in service and it was not until 1915 that a solution of the problem was reached by the development of the M.A.N. four cycle engine to which I have previously referred.

Since the war, the other European designers, viz.: Vickers-Armstrongs, Tosi, Fiat, Sulzer and Schneider have worked to bring their product up to the M.A.N. 1915 standard as to weight and reliability. No new principle of construction or operation has been applied to this development by any of the designers just named, except Vickers-Armstrongs. The objective has been sought and attained solely by refinements in design, the use of special high grade materials, etc. The Vickers-Armstrongs development is distinguished from the others named by the fact that that firm has continued to develop and apply their system of mechanical fuel injection while the others, except M.A.N., have adhered to air injection. As the result of all this development, there are now available in Europe, in addition to the M.A.N. types, *six different designs* of single acting submarine engines. Of these, three are four cycle, viz.: Vickers-Armstrongs, Tosi and Schneider and three are two cycle, viz.: Sulzer, Fiat and Schneider-Carels.

Before leaving this part of the subject, it may be of interest to point out that the increased dimensions of some of the post-war European submarines afford the necessary headroom for the installation of crosshead instead of trunk piston engines and accordingly the crosshead type appears in some of the latest constructions.

Simultaneously, M.A.N. have been engaged both in refining and improving their four cycle, single acting engine and in developing a high speed, double acting, two cycle engine.

The changes made by M.A.N. in their latest (1929) four cycle, single, acting, trunk piston engine include the elimination of the air compressor and the substitution of mechanical, or "solid," injection. The injection system employed, however, differs materially from that employed on the Vickers-Armstrongs engine and is the same as that used for some years past by M.A.N. and the Electric Boat Company on commercial engines. As compared with the 1915 type, a saving in weight of about 42 per cent has been achieved. Elimination of the air compressor accounts for about 14 per cent, and increase in piston speed for about 12.5 per cent, and the balance is accounted for by the use of an increased number of cylinders of smaller dimensions combined with refinements in design.

The port scavenged, double acting, two cycle, mechanical injection commercial engine is now a fully demonstrated success in the marine and stationary fields. This development, which started with heavy, slow speed, air injection engines, has culminated in the very light weight, mechanical injection engine constructed by M.A.N. for the propulsion of German cruisers. In this connection, it is interesting to note that all of the leading Diesel engine builders in Europe have adopted or are experimenting with the double acting, two cycle system for large commercial engines.

As compared with all single acting, non-supercharged engines, the double acting, two cycle, mechanical injection engine offers inherent and very obvious advantages with respect to the weight and space required for a given output at given revolutions when constructed to meet the same standards as to reliability and life. For submarine service, where space and weight available are both limited, comparatively high revolutions are absolutely essential and accordingly the application of the double acting, two cycle principle to submarine engines had to wait until it had been demonstrated by actual tests that engines of this type could be successfully operated at high revolutions. The M.A.N. construction of such engines for commercial purposes has been marked by a continual increase in revolutions with a corresponding decrease in weights. For naval purposes, including submarines, lighter engines with still higher revolutions are necessary. The information necessary for the successful design of such engines has been gained by M.A.N. from the construction of commercial engines, experimental engines and the German cruiser engines. Simultaneously, similar data has been accumulated by the Electric Boat Company by the construction and testing of commercial and experimental engines.

While it has been fully demonstrated that the mechanical injection principle can be successfully applied to much larger double acting, two cycle cylinders than would ever be required in submarine engines, there appears to be a lower limit of cylinder dimensions below which this construction is not practicable, mainly on account of the difficulties encountered in securing satisfactory combustion (and satisfactory exhaust) on the lower side of the piston. These difficulties arise from the small dimensions of the

annular combustion chamber. While it has been demonstrated that with air injection somewhat smaller double acting submarine engine cylinders are practicable, the inferiority of air injection to mechanical injection would appear to rule out such engines and accordingly the minimum diameter for the double acting, submarine cylinder is now thought to be 14 to 15 inches, the minimum corresponding stroke being substantially 19 inches.

Except for refinements in design and the use of higher mean effective pressures and piston speeds, the reductions which have been achieved in the weight and dimensions of single acting engines per horsepower delivered are ascribable to the substitution of mechanical, or "solid" injection, for air injection. This change from air to mechanical fuel injection results in simplification of design, reduction in length, weight and fuel consumption, better balance, increased reliability and decreased maintenance.

Since the double acting, two cycle, mechanical injection engine combines minimum length and weight with the necessary degree of reliability and durability, it seems destined to eventually displace the single acting engine within the power range for which it is practicable and suitable. Accepting four working cylinders combined with two scavenging cylinders as the minimum consistent with the balance required for high speed and accepting 14 to 15 inches as the minimum cylinder diameter for such engines, the minimum power is in the neighborhood of 1500 B.H.P. and this power output combined with the height required indicates that the field for such engines will be restricted to submarines of approximately 1500 standard tons displacement and up.

This leaves a standard displacement range of from 600 to approximately 1500 tons where the use of single acting engines is obligatory. Owing to its inherent and fully demonstrated advantages, the mechanical fuel injection system seems destined to replace the air injection system on all such engines. It would therefore appear that the lower power range in question will, in the future, be covered by two main types, viz.: the single acting, trunk piston, mechanical injection engine, which may be either two or four cycle, and the supercharged, single acting, four cycle, crosshead, mechanical injection engine.

Crosshead engines have many advantages:—

First—Increased reliability by reason of accessibility of all important bearings while engine is running. This carries with it not only much greater freedom from casualties occasioned by inefficient lubrication but the further advantage of greatly minimizing consequential damage and time, cost and effort required to make good such casualties.

Second—Simplification and increased reliability of piston cooling apparatus.

Third—Elimination of side thrust on pistons with reduction in cylinder wear.

Fourth—Decrease in lubricating oil consumption of about 65 per cent.

Fifth—Lubricating oil consumption substantially independent of wear, carrying collateral advantages as to control of cylinder lubrication and clear exhaust.

Sixth—Elimination of contamination of crankcase oil, carrying with it increased reliability.

Seventh—Decrease in lubricating oil storage space and weight with corresponding increase in fuel oil capacity and radius of action.

The headroom available in any particular case will, of course, fix the maximum stroke of the engine and this, in turn, within certain limits will fix the diameter. Since undue multiplication of cylinders would result in excessive length, the crosshead type would not appear to be suitable for small boats where the conditions appear to force the continued use of trunk piston engines.

Since at constant displacements, speed and hull dimensions may and do vary, it follows that a belt rather than a line will divide the natural fields of the three types of engine under discussion.

A very cursory investigation indicates the following as the natural fields for the three types, the overlaps in powers and displacements representing the belts above referred to:

ENGINE FIELDS.

Type of Engine	Standard Displacement Tons	Total Engine Power
Single acting, trunk piston, 2 or 4 cycle, mechanical injection . . .	Up to 600	Up to 1400 B.H.P.
Single acting, crosshead, 2 or 4 cycle, mechanical injection	450-1700	900-5000 B.H.P.
Double acting, 2 cycle, mechanical injection . .	1500 up	4500 B.H.P. up

All Diesel engines for submarines have somewhat the same requirements in common:

1. Greatest possible power for the least possible weight and space, which for convenience we may call the rating of the engine.

2. Ability to occasionally develop 10 per cent more than this, which is the maximum or overload rating.

3. Ability to run continuously at some lesser percentage (preferably small) than the full power rating.

The above three requirements always assume good maintenance and operation.

4. Small fuel consumption per I.H.P. The smaller this is originally, the lesser the maintenance required and the longer the engine can operate under emergency conditions before overhaul.

5. Small lubricating oil consumption. This is important because the smaller amount used, the lesser is the amount of residue required to be taken out of valve cages, passages, lubricating oil, etc., and also the smaller quantity required to be carried in the storage tanks.

Both 4 and 5 are important because they bear on the radius of action of the submarine.

6. Renewable wearing parts, that is—Cylinder liners must be fitted; valves should be in cages; all bearings adjustable; cylinder heads should be as simple as possible; no part should be of excessive weight; every wearing part must be accessible for routine

inspection, maintenance and repair; while it is important that all reciprocating parts should be as light as possible.

7. Perfect balance as regards primary and secondary forces and couples.

8. The general design to be such that the major critical speeds of the whole system shall not occur in the operating range of the engine, or means provided to keep the engine isolated from the remaining part of the propelling system insofar as lengths of shafting and rotating masses are concerned, but connected thereto in the sense that power may be transmitted. This means an elastic coupling of some description, because then one only has to deal with the critical speeds of the engine itself and not with the engine plus the combinations caused by surface and charging conditions.

Some judges believe that absolute smokelessness of exhaust under all conditions is absolutely essential. Any well designed engine, air or mechanical, will comply with this requirement when kept in good condition and supplied with suitable fuel. Conversely any engine with either type of injection will show smoke under some conditions, such as when due for routine overhaul, using the dregs of fuel tanks when overloaded in some one or two cylinders through maladjustments on the others, and I believe, even for S/m service the requirement is not so important as generally believed. A S/m when exerting maximum power in a chase, for instance, will not care much if the engines do smoke under such conditions. If proceeding to a distant rendezvous under easy power, the engines will not emit smoke, even though needing some adjustment. If pursued, the vessel must dive. The clearness of exhaust is mostly a reliable indication that the upkeep of the machinery has been sufficient, given suitable fuel oil, which our S/m always have supplied them. A Senior officer of a flotilla, therefore, has a very good guide as to the relative efficiency of the machinery in any group of sister vessels performing the same type of duty.

It is of course highly important that adequate time be allotted for routine examination and maintenance, and it is a criterion of engineering ability to so provide for such examination, etc., that wear and adjustments never become casualties. In other words, prevention is better than cure. Such a routine, once estab-

TABLE I.

SUBMARINE ENGINES.

Design of Sub. used on		Type of Engine	Type	No. of Cylinders and Power	Date of Design	Bore and Stroke	R.P.M.	M.E.P.	Lbs. per B.H.P.	Piston Speed Ft/Min.
E. B. Design	U. S. Class									
		Gas				Inches				
7A to G	A	4 cycle	Otto	4/160	1900	11½ × 14	350	65	87.1	815
16–A	B	4 cycle	Craig	6/250	1905	11 × 12	500	58	52.2	1000
17A and C	C	4 cycle	Craig	6/250	1907	11 × 12	500	58	52.2	1000
12–A	D	4 cycle	E. B.	6/300	1907	12 × 12	500	58.5	63.5	1000
19–A	E	Diesel	Vickers	4/275	1909	12½ × 13½	400	79	64.1	900
20–B	F	4 cycle	Vickers	6/390	1910	12½ × 13½	400	74.6	67.4	900
26–B	H	2 cycle	E. B.	6/450	1911	11 × 12½	450	55.6	63	920
30–B	K	2 cycle	E. B.	6/450	1912	11 × 12½	450	55.6	63	920
32	K	2 cycle	E. B.	6/450	1913	11 × 12½	450	55.6	63	920
37–Ga	L	2 cycle	E. B.	6/450	1914	11 × 12½	450	55.6	63	920
39–E	M–1	2 cycle	E. B.	6/420	1915	10½ × 16	350	57.2	99	930
61–B	N	4 cycle	E. B.	8/240	1915	9 × 12½	375	80	97.7	780
68–A	O	4 cycle	E. B.	6/440	1916	13½ × 14	400	72.4	79.5	930
77–A	R	4 cycle	E. B.	6/440	1916	13½ × 14	400	72.4	79.5	930
73A and C	S18/47	4 cycle	E. B.	8/600	1917	13½ × 15	380	72.8	89	950
EB–16	S3	4 cycle	E. B	8/700	1916	14½ × 16	350	75	89.3	935
63A and C	T	4 cycle	E. B.	6/1000	1916	18 × 19	375	73	81.1	1190
SV 35/35	....	4 cycle	M.A.N.	6/550	1916	13.78 × 13.78	450	78.5	60	1035
SV 45/42	....	4 cycle	M.A.N.	6/1200	1915	17.72 × 16.54	450	86.3	45.3	1240
SV 53/53	....	4 cycle	M.A.N.	6/1750	1916	20.86 × 20.86	380	85.5	55	1320
Schneider	....	2 cycle	Schneider	8/1500	1928	16.15 × 17.75	330	61.8	61.0	975
Fiat	....	2 cycle	Fiat	6/1500	1928	16.54 × 16.54	420	66.7	51.3	1158
Tosi	....	4 cycle	Tosi	6/1500	1928	20.87 × 20.87	370	74.8	95.5	1285
Vickers-Armstr.	....	4 cycle	Vickers	6/1500	1928	20 × 19½	400	81.8	55.3	1282

lished enables the C in C to know that at any given time, a certain proportion of his fleet is capable of certain definite action, without undue exertion or strain on the personnel or repair facilities, and that such degree of preparedness can be maintained for the maximum time, and also to know that for the real emergency every unit would still have something in reserve.

Table I gives the characteristics of the majority of the engines fitted in U. S. submarines up to the S. Class, with a few representing modern European practice. Finally, it cannot be too strongly impressed on submarine engine designers that it is not a case of just designing a Diesel engine—many other considerations carry much weight, the harmonious design of the whole boat being the major quest, and it must not be forgotten that U. S. and European standards differ considerably and one may cite the example of the M.A.N. two cycle stepped trunk piston engine fitted in H-1/3, K-1/8, L-1/4 and L-9/11. This design of engine was considered eminently satisfactory in a number of European navies long after it had been abandoned by the U. S. Navy.

LIGHT WEIGHT DIESEL ENGINES.

By O. D. TREIBER, MEMBER.

The weight per horsepower of Diesel engines has commonly been rather high since this form of prime mover was first built, ranging from 150 to 600 pounds per horsepower with few exceptions running as low as 60 pounds per horsepower.

To reduce these weights, we have three principal methods which are:

1. Using higher strength materials and stressing them higher.
2. Increasing the speed of revolutions.
3. Increasing the mean effective pressure.

A Diesel engine is comparatively simple. Insofar as construction is concerned, the problems involved in the first and second division of this study are generally the same as those involved in gasoline engine construction.

In the first item of materials, we have a great variety to choose from, depending on the type, size, service and cost of an engine. The metallurgists' contribution of metals gives us many metals available now which were not to be had until recent years or months. We have, for instance, steel castings in thin intricate sections which could not be made up to a short time ago. Steel castings can be stressed many times higher than cast iron with equal safety. Heat treated aluminum forgings and castings contribute their portion to reducing weight. In addition to these two great factors for reducing weight, we have many steel alloys, heat treated, which can be stressed two or three times as high as plain carbon steel, which was the principal metal used up to a few years ago.

Cast iron of a decade ago had normally a tensile strength of 22,000 pounds per square inch and 2000 pounds was considered a top safe limit for stressing in engine construction, while today we have cast nickel iron and semi-steel of 35,000 to 40,000 pounds

tensile strength. Furthermore, we have iron today 25 to 60 points Brinell harder than was available a few years ago.

We have improved bearing metals and have a better knowledge of bearing construction and lubrication today that permits much higher bearing loads with equal safety. The roundness and polish on journals also contributes immeasurably to increased bearing loads with safety. With higher bearing loads we can get cylinder centers closer together and eliminate a great deal of weight. Cylinder centers are no more fixed by length of bearing required but by the cylinder liners and head construction. Increasing revolutions decreases maximum bearing pressures up to the speed at which the inertia forces of reciprocating parts equals one-half the maximum gas pressures. Main bearing loads are reduced by using counterweights on the webs of the crankshaft but of course add to the weight of the engine. Maximum bearing loads are of greater importance to load limits than mean pressures.

By utilizing these improved metals and eliminating metal in an engine except where it is required, we can greatly reduce the weight per unit of cylinder volume.

Reducing the weight of an engine per horsepower by increasing the revolutions involves the problems of balance, stresses from inertia and centrifugal forces, gas velocity through inlet ports or valves (which is one of the most important factors limiting revolutions), time lag of ignition and distribution of fuel.

Gasoline engines have been run over 20,000 R.P.M. Many as high as 10,000 R.P.M. and in racing cars 7000 to 8000 R.P.M. is done. These higher speeds, of course, are attained with forced induction.

The time lag of ignition and distribution of fuel offer one of the principal problems to master in the high revolution Diesel. The fuel being injected into the hot compressed air in the cylinder has to be raised in temperature from that at which it is injected to the temperature at which it will unite with the oxygen in the air. Obviously the higher the temperature at the starting of injection, the less time it will take to raise it to the temperature of chemical union with the oxygen but if raised to too high a temperature the oil will decompose and clog up the injection devices. Furthermore, the coarser the spray or particles of fuel after leaving the nozzle the greater the time required to either gasify these

particles, or droplets, or for the oxygen to absorb the hydrogen and carbon molecules from the surface of the droplets until they are completely absorbed. It is, therefore, obvious that these droplets should be as small as possible to reduce the time required for their complete uniting with the oxygen to form CO_2 and H_2O. The uniting of these elements generates heat and heat causes expansion or the power creating force that drives the piston downward. The crankshaft of an engine running at 1200 R.P.M. turns 7200 degrees per second or .00014 seconds per degree.

The lowest time lag of ignition with fine spray is about .0007 of a second which would be the equivalent to 5 degrees of crankshaft travel. The time lag is reduced as the temperature of the cylinder gases increases. If 30 per cent of the oil is burned at constant volume or at top center, the pressure will be approximately twice the compression pressure which is considered about the limit for practical constructional reasons. Furthermore, the thermal efficiency is not materially increased by burning more at top center, but must be burned as soon after top center as possible. So it would appear that 30 per cent of the oil should be injected prior to 5 degrees before top center and 70 per cent between 5 degrees before center and at a rate to continue the pressures reached by the burning of the 30 per cent at top center until all of the fuel has united with the oxygen to form CO_2 and H_2O. If the cylinder were filled to 100 per cent capacity with air, and combustion were complete, using all of the oxygen in the air, the injection would be completed at 30 degrees past top center allowing 5 degrees for lag and in which case the mean pressures would be approximately 190 pounds. However, this is not possible as there must be some excess oxygen and 100 per cent volumetric efficiency is not readily possible with natural asperation. Furthermore, the carbon does not all burn to CO_2. There are other problems to consider such as disassociation, sulphur in the fuel, moisture, etc. Other limitations of constructional difficulties further limit the high M.I.P. which, if accounted for, would indicate that 170 pounds M.I.P. would be the theoretical limit with a thermal efficiency of 42 per cent. One test record has shown 160 pounds M.I.P. with 33 per cent thermal efficiency, another 115 pounds M.I.P. with 34.5 per cent thermal efficiency and another 95 pounds M.I.P. with 38 per cent thermal efficiency.

The attaining of high M.I.P. involves not only the proper spraying of fuel into the combustion chamber but brings it in contact with the molecules of oxygen in the air in time to unite and give off its portion of heat before the expansion ratio drops. To accomplish this function more perfectly a turbulence of the air in the combustion chamber often assists materially and many different types of combustion chambers have been designed for the purpose of bringing the fuel and oxygen in the air together more completely and rapidly.

Devices for spraying fuel into these combustion chambers are very numerous.

This outlines briefly the general problems of light weight Diesel engines. They can be built almost as light per unit of piston displacement as a gasoline engine in reasonably good size cylinders and turn about as fast, but mean indicated pressure as high as some gasoline engines develop has still to be done consistently.

We have built a number of light weight Diesel engines ranging from 65 horsepower to 2500-3000 horsepower for marine and railroad service, weighing from 12 to 30 pounds per horsepower. Lighter engines can, of course, be built at somewhat greater cost and the revolutions per minute of many of these engines can be increased to give higher power and thus reduce the weight. However, for practical service and in keeping with consistent foundry limitations these engines are as light as may be reasonably expected for the service they are to perform.

THE U. S. S. *LOUISVILLE* AND U. S. S. *CHICAGO* STEERING GEARS.

By LIEUT. (J.G.) J. F. COOKE, U.S.N.R., MEMBER.

In view of the fact that Scout Cruiser No. 28, U. S. S. *Louisville,* and Scout Cruiser No. 29, U. S. S. *Chicago,* will join the United States Fleet sometime during 1931, a general description of the steering gears installed on these ships will no doubt be of interest to naval personnel ashore and afloat.

The principal parts of the steering gears and the primary function of each are tabulated as follows (see Fig. I):

(*a*) *Crosshead*—Horizontal member of the steering gear proper, directly connected to the rudder stock, which transmits any movement of the rams to the rudder.

(*b*) *Cylinders*—Cylindrical chambers in which the rams move, due to the oil pressure received from the size L.P. 360 "Hele-Shaw" hydraulic pumps.

(*c*) *Connecting Links*—Links connecting rams to crosshead.

(*d*) *"Follow Up" System*—That part of the steering gear which returns the valve spindle of the hydraulic control cylinder, through the mechanical differential, to the neutral position allowing the pressure from the P. 75 "Hele-Shaw" hydraulic pump to return the pump control spindles of the L.P. 360 "Hele-Shaw" hydraulic pump to the neutral position, thus stopping any pumping action to the cylinders until another movement of the rudder is desired.

(*e*) *Trick Wheel*—Steering wheel located adjacent to the steering gear. Used for testing steering gear and for steering from steering gear compartment.

(*f*) *Power Units*—2—L.P. 360 "Hele-Shaw" variable stroke, variable capacity, hydraulic pumps, each driven by a General Electric Company motor (75 H. P., 230 volt, 850 R. P. M., full load, 1000 R. P. M., light load; direct current, shunt wound) directly connected through gearing. These pumps deliver oil to the cyl-

inders and the direction of the discharge and rate of delivery is determined by the control mechanism.

(*g*) *Control Mechanism*—One control mechanism is required for each power unit. This mechanism consists of a P. 75 "Hele-Shaw" variable stroke, variable capacity, hydraulic pump, a hydraulic control cylinder, and a bevel gear mechanical differential equipped with "follow-up" and control connections.

(*h*) *Replenishing Tank*—An oil reservoir provided with connections to both sides of the system through a shuttle valve. This tank is continuously in direct connection with the suction side of the system, which is therefore filled at all times.

(*i*) *Drain Pumps*—2—Gear type pumps, driven by sprocket and chain drive from the main motors. When the main motors are secured these pumps are driven by a ½ H.P., 1120 R.P.M., direct current, Star Electric Motor Company motor. These pumps are used to drain the L.P. 360 "Hele-Shaw" pump bedplates and to return this oil to the expansion tank. They are also used to drain the entire system when necessary.

(*j*) *Hydraulic Torque Equalizer*—An automatic safety device, actuated by the pressure in the main system, connected to one end of the valve spindle of the hydraulic control cylinders, and operating when the pressure in the main system is equivalent to 150 per cent of the normal horsepower rating of the main electric motors. The pre-determined pressure cannot be exceeded in the main system as this device will function and return the L.P. 360 "Hele-Shaw" pump to neutral stroke, thereby stopping pumping action.

(*k*) *Valves*—4—Stop valves, 2—by-pass valves, 2—shuttle valves, 4—check valves, 4—oil drain valves.

(*l*) *Electric Control System*—1.—3 Forward or Transmitter General Electric Selsyn Motors.

2.—2 After or receiver General Electric Selsyn Motors.

3.—Collector rings mounted on commutator end of main electric motor armature shafts to supply alternating current for Selsyn motor system.

4.—2 Transfer panels for port and starboard cable leads from Central Station to steering gear compartment.

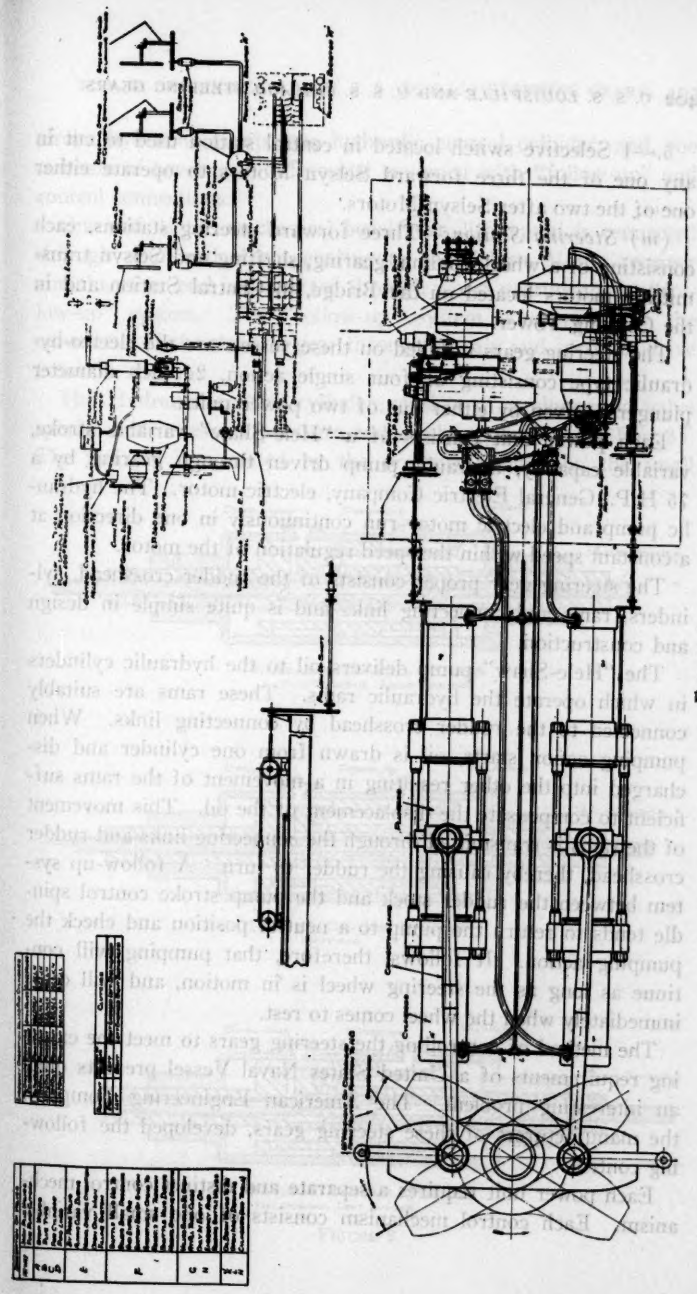

FIGURE 1.

5.—1 Selective switch located in central station used to cut in any one of the three forward Selsyn Motors to operate either one of the two after Selsyn Motors.

(*m*) *Steering Stations*—Three forward steering stations, each consisting of a wheel, column, gearing, shafting and Selsyn transmitting motors located on the Bridge, in Central Station and in the Conning Tower.

The steering gears installed on these vessels are the electro-hydraulic type consisting of four single action, 24 inch diameter plungers, driven by either one of two power units.

Each power unit consists of a "Hele-Shaw" variable stroke, variable capacity, hydraulic pump driven through gearing by a 75 H.P., General Electric Company, electric motor. The hydraulic pump and electric motor run continuously in one direction at a constant speed within the speed regulation of the motor.

The steering gear proper consists of the rudder crosshead, cylinders, rams and connecting links and is quite simple in design and construction.

The "Hele-Shaw" pump delivers oil to the hydraulic cylinders in which operate the hydraulic rams. These rams are suitably connected to the rudder crosshead by connecting links. When pumping action starts, oil is drawn from one cylinder and discharged into the other resulting in a movement of the rams sufficient to compensate the displacement of the oil. This movement of the rams is transmitted through the connecting links and rudder crosshead, thereby causing the rudder to turn. A follow-up system between the rudder stock and the pump stroke control spindle tends to return the pump to a neutral position and check the pumping action. It follows, therefore, that pumping will continue as long as the steering wheel is in motion, and will cease immediately when the wheel comes to rest.

The method of controlling the steering gears to meet the exacting requirements of a United States Naval Vessel presents quite an interesting problem. The American Engineering Company, the manufacturers of these steering gears, developed the following control.

Each power unit requires a separate and distinct control mechanism. Each control mechanism consists of one small variable

stroke hydraulic pump, one hydraulic control cylinder, and one mechanical bevel gear differential equipped with "follow-up" and control connections.

One end of the mechanical bevel gear differential is connected through spur and worm gearing to the Selsyn Motor receiver and the other end is connected through worm gearing to the "follow-up" system. The "follow-up" system is operated by the movement of the steering gear through a chain and sprocket drive from the rams.

The Hydraulic control cylinder consists of a plunger, a valve spindle and necessary pressure and discharge ports (see Fig. 2). The plunger is operated by the pressure created by the small

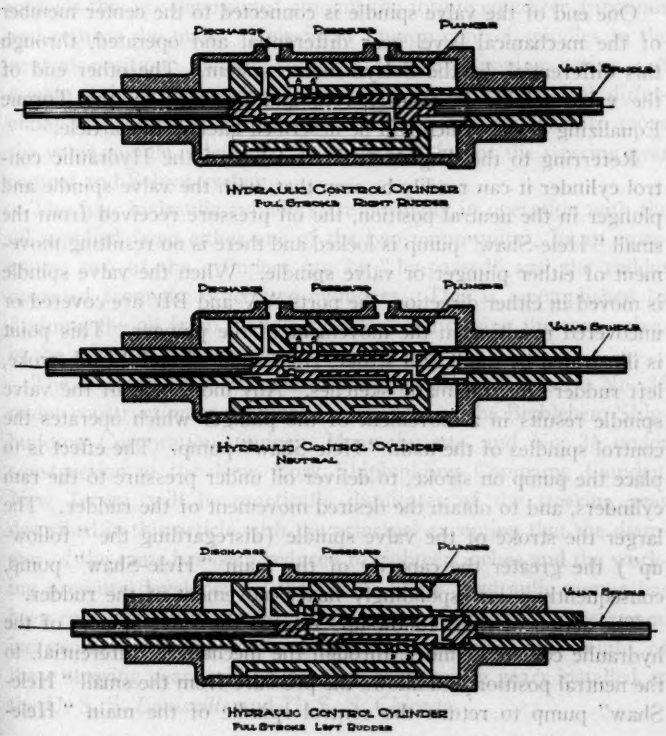

FIGURE 2.

" Hele-Shaw" hydraulic pump. This pump is similar in construction to the main " Hele-Shaw" steering gear pump and is driven through sprocket and chain from the main electric motor or, it may be driven by the ½ H.P. electric motor used to drive the gear type drain pumps.

The small " Hele-Shaw" pump is equipped with a pressure regulating device, which can be set for a predetermined pressure, which controls the movement of the pump control spindle and allows the pump to deliver oil to the hydraulic control cylinder at a limited maximum pressure. The plunger of this cylinder is connected through levers to the control spindles of the main " Hele-Shaw" pumps.

One end of the valve spindle is connected to the center member of the mechanical bevel gear differential and operated, through this differential, by the Selsyn motor system. The other end of the valve spindle is connected to and operated by the Torque Equalizing device which will be described later in this article.

Referring to the diagrammatic sketches of the Hydraulic control cylinder it can readily be seen that, with the valve spindle and plunger in the neutral position, the oil pressure received from the small " Hele-Shaw" pump is locked and there is no resulting movement of either plunger or valve spindle. When the valve spindle is moved in either direction, the ports AA′ and BB′ are covered or uncovered resulting in the movement of the plunger. This point is illustrated by the " Full stroke, right rudder" and " Full stroke, left rudder" diagrammatic sketches. Any movement of the valve spindle results in a movement of the plunger which operates the control spindles of the main " Hele-Shaw" pump. The effect is to place the pump on stroke, to deliver oil under pressure to the ram cylinders, and to obtain the desired movement of the rudder. The larger the stroke of the valve spindle (disregarding the " follow-up") the greater the capacity of the main " Hele-Shaw" pump, consequently a correspondingly faster movement of the rudder.

The action of the " follow-up" returns the valve spindle of the hydraulic control cylinder, through the mechanical differential, to the neutral position and allows the pressure from the small " Hele-Shaw" pump to return the control spindle of the main " Hele-

Shaw" pump to the neutral position, thus stopping the pumping action until another movement of the rudder is desired.

The hydraulic torque equalizing device is operated through a shuttle valve by the oil pressure in the main system when the pressure reaches the equivalent of 150 per cent of the normal horsepower rating of the main electric motor. This device consists of two single action plungers, the plunger cylinders connected through a shuttle valve to the high pressure side of the hydraulic system. As long as the pressure in the system does not reach this predetermind amount the torque equalizer operating lever, which is connected to the hydraulic control cylinder spindle, moves freely between the plungers. When the pressure in the main system exceeds this pre-determined amount the torque equalizer functions and returns the main "Hele-Shaw" pump control spindles to the neutral position, thereby stopping further pumping action. A double action compression spring between the mechanical differential and the valve spindle permits the torque equalizer to move the valve spindle independently of the position of the steering gear control and Selsyn system.

The four hydraulic cylinders are normally in operation with the oil supplied from either one of the two pump units. In an emergency, two of the cylinders can be "by-passed" and the rudder operated from the other two cylinders. This is accomplished by draining the oil from the idle cylinders and closing the stop valves in the pressure supply pipes.

The steering gears for the most recent Scout Cruisers, No. 33 under construction at the Fore River Plant of the Bethlehem Shipbuilding Corporation, Quincy, Massachusetts, and No. 35 under construction at the New York Shipbuilding Company, Camden, New Jersey, will be practically duplicates of the steering gear described in this article with the principal exception that the diameter of the rams has been reduced to eighteen inches and the working pressure developed by the "Hele-Shaw" hydraulic pumps has been considerably increased. By decreasing the ram diameter and increasing the pump working pressure, the size and weight of these steering gears will be less than the steering gears installed on the U. S. S. *Louisville* and U. S. S. *Chicago*.

NEW TYPE GYRO AND LIGHT LUBRICATING OIL.

By Lieut.-Comdr. Roy C. Smith, Jr., U.S.N.

HISTORICAL.

The question of a suitable oil for the lubrication of torpedo gyroscope bearings is probably as old as the use of gyros in torpedoes. The requirements of a proper oil for this purpose are quite definite. It must be at once an excellent preservative and an excellent lubricant. It must adhere closely to the metal parts, and must not wash off when submerged, nor be blown off by air blasts. It must not gum, nor harden on the parts when standing for long periods in storage, and the gyro should be ready for a shot when taken from storage without the necessity of cleaning or re-oiling the bearings.

Various oils have been tried with greater or less degree of satisfaction. Apparently the earliest oil used was sperm oil. Difficulties arose with this oil in the way of obtaining a satisfactory acid free article, and in the fact that it turns rancid in time. These disadvantages caused various other oils to be tried, which are not now a matter of accessible record. At length, about 1912, an investigation was made into the general question having regard to the practice of other makers and users of various light and delicate machinery, such as clocks and watches. As a result, porpoise jaw oil was adopted for gyro bearing use.

This oil was apparently satisfactory in service until November, 1920, when the Torpedo Station discovered that some oil received and issued in pint cans contained an excessive amount of acid, 4.5 per cent, and was not fit for use in gyro bearings. A number of gyro bearing cups and centers were returned from Submarine Base, New London, in poor condition, pits were quite noticeable and around the brass retainer ring verdigris showed.

At about the same time, the E. W. Bliss Company was furnishing a special oil (probably porpoise jaw) with gyros, and various

other oils were recommended. Accordingly, tests were made for acidity and results obtained as follows:

	Per Cent Acid
No. 1 Oil furnished by E. W. Bliss	.73
No. 2 Porpoise jaw (pint cans), Wm. F. Nye Co	4.5
No. 3 Sperm oil (previously abandoned)	.04
No. 4 Porpoise jaw (gallon cans, source unknown)	.10
No. 5 Rotor bearing oil (Sperry Co.)	Neutral.

This last oil was furnished bythe Vacuoline Oil Company and is known under the trade name Vacuoline Extra "B." As a result of the above, the use of porpoise jaw was discontinued at this Station, and the Sperry rotor bearing oil (Vacuoline) adopted. A recommendation to this effect to the Bureau was approved and on 30 November, 1920, the Bureau issued a circular T-17, ordering the change made.

In March, 1921, the Naval Inspector of Ordnance at E. W. Bliss Company requested permission to use porpoise jaw oil again, as it was not thought that Bliss could pass gyros with the Vacuoline oil due to its greater viscosity. The Station replied that there was no appreciable difference in the performance with the two oils, but that the acid in porpoise jaw oil caused pitting of the balls. Upon request of the Bureau, this Station conducted tests of spinning gyros with both oils. Four Mark IX gyros were used and results showed from three to five minutes longer spin with the porpoise jaw oil. In July, 1921, the Commander, Submarine Base, San Pedro, requested permission to use porpoise jaw oil in place of Vacuoline, as the latter washed off from torpedoes which had been fired and not immediately recovered. Within eight hours, in such cases, there is formed a thin coating of rust on pivots and bearings. Porpoise jaw oil sticks longer and prevents this.

About August, 1921, the E. W. Bliss Company again objected to the use of Vacuoline in the gyro bearings, apparently not considering it sufficiently sensitive nor a satisfactory preservative. They recommended the trial of E. W. Houghton Company's Rust Veto products for this purpose, and the Torpedo Station wrote to that company explaining its problem and asking recommen-

dations. The E. W. Houghton Company replied, offering their "Rust Veto Amber Liquid 'V'," which is a pure mineral oil mixed with a small percentage of potassium chromate to give preservation and adhesive qualities lacking in the mineral oil. This product was extensively tried out in service and reported satisfactory. It was recommended to the Bureau as acceptable, and its use was ordered in July, 1922. Shortly, complaints began to be received that this Amber Liquid "V" was too viscous to give the necessary degree of sensitivity, and that it had a high pour point. After consultation with a salesman for the E. W. Houghton Company, their product "Rust Veto No. 40," a slightly different compound, was accepted by the Torpedo Station, and in April, 1925, a change to this oil was recommended to the Bureau.

There is no record of complaint with Rust Veto No. 40 until December, 1926, when it was reported that when the oil was allowed to stand, separation took place. Examination of a gyro which had been laid up for a considerable length of time showed the bearing to be discolored, and a varnish like ring of dried oil around each ball. The bearings could not be cleaned in the usual manner—washing in gasoline and blowing off—but had to be polished in a machine in preparation for issue. Various other bearings, lubricated with Rust Veto products, showed similar sticking, and reports received from U. S. S. *Melville*, *Whitney*, *Dobbin*, and Battle Fleet Torpedo School at San Diego all showed that after a very short time the oil (in this case Rust Veto No. 40) "congeals and binds the balls like glue. It is an excellent preservative but causes the gyro to be sluggish even though the balls are free." This condition caused the Torpedo Station to abandon use of Rust Veto products and to return to the Vacuoline Extra "B" previously used, and in August, 1927, the change back was approved by the Bureau.

All these conditions caused the Station to continue the research problem, and various other oils and compounds have been tried out with a view to finding something that would be as good a lubricant as the Vacuoline and as good a preservative as the Rust Veto products. Any lead was followed that seemed to promise any advantages, but nothing of value was found out, beyond that the Vacuoline was the best mineral oil we could get, and the Rust Veto

Amber Liquid " V" was excellent except for the tendency to gum. About this time, more trouble with rusty bearings was had on the range, and for Station work the use of Vacuoline was abandoned, and the Amber Liquid " V" restored.

In April, 1928, a suggestion was made by the Engineering Experiment Station, Annapolis, Md., that a little acidless tallow oil mixed with the Vacuoline might give that oil the necessary adhesive qualities which it lacked. This problem was referred by the Research Officer, Lieutenant Commander R. W. Christie to Dr. A. H. Gill, of the Massachusetts Institute of Technology, one of the leading authorities on the subject, and to Dr. F. R. Baxter, of the Vacuum Oil Company. Dr. Gill thought that the addition of this oil might help, or that the addition of a quantity of pure neatsfoot oil might help. Dr. Baxter did not recommend the mixture. Both men, however, said that porpoise jaw oil was what our problem called for and expressed surprise that we had had trouble with it, Dr. Gill stating: " This oil has been used in watches and the very finest machines for over forty years and is still so used. No appreciable deleterious effect has resulted from its use."

This brought the investigation back to the situation in 1912, when porpoise jaw oil was first adopted. Letters were written to various clock and watch manufacturers and all were found to use porpoise jaw oil or an oil of which it was the major constituent. It was also learned that makers of automatic alarm systems used porpoise jaw in their machines, having found it the only safe one to use for just such a problem as ours.

The supply of this oil was known to be very small and precarious. It was found that some 200 gallons of gyro oil had been issued annually from this Station. The only two producers of porpoise jaw oil were consulted, and both the Wm. F. Nye Co., of New Bedford, Mass., and the D. C. Stull Co., of Provincetown, Mass., replied stating that the figure was quite out of the question. There wasn't that much oil. Our requirements were reviewed and a casual guess of 50 gallons a year was made, allowing for lubrication of gyros only, but this also was out of the question. Mr. Howard I. Wordell, of the Nye Company, visited the Station and went thoroughly into the problem of production and refining with us. From records which he submitted of his oils in service,

and from further correspondence with Dr. Gill, the conclusion was arrived at that porpoise jaw oil was the oil we wanted, and that it only, would do what we required. Samples of their oil were procured in March, in May, July and August, and these samples gave every indication of being entirely satisfactory for our purpose. From our investigation, we came to the conclusion that the original trouble with this oil, acidity, was accidental and with that careful test of all oils received there should be no danger of issuing again any oil with too high an acid content.

The cycle was now complete, and we were back where we were in 1912. We had returned to the realization that pure porpoise jaw oil was the only pure oil suitable for our purpose, which belief other users of this oil never abandoned.

ABOUT PORPOISE JAW OIL.*

Ever since men began to go down to the sea in ships they have had to pin their faith upon the reliability of chronometers. Any failure of the lubrication involved serious errors of navigation. The whaling industry of New Bedford, whose ships were often three and four years on a voyage, felt this unreliability perhaps more than any other class of time-piece users. It was in response to their demand for reliable oil, that one of the oldest manufacturers of chronometers in the Whaling City discovering that an oil properly prepared from the jaw of a porpoise gave the most absolutely perfect results, commenced its manufacture. In those days horological instruments were far from common, and it was only the favored few who carried time-pieces. The crude article brought in by whale ships to be refined was at first ample in quantity, but with the rapid progress of the horological art there came a larger demand, and stronger inducements were held out to the New Bedford whalers to bring in greater quantities of this Fish Jaw Oil. The increasing demand and consequently higher prices offered, induced much search for a substitute, and many kinds of lubricators were tried. Nut Oil—Bone Oil—Seed Oil—Mineral Oil, all in their turn were tried, but proved far less reliable for the purpose.

* Much of this information was supplied by the Wm. F. Nye Co., New Bedford, Mass.

The perfect lubricator does not oxidize, grow rancid, evaporate, creep, gum or chill under a low temperature. For seventy years, since fish jaw oil has been known, nothing has proved its equal in all these requirements, when properly prepared. With the rapid increase of time-pieces, came a most strenuous demand for the crude material, and while this demand grew greater, the whale fishery declined, and production grew gradually less.

Again, for many years, the obtaining of this oil was in the hands of American whalemen, whose reliability could not be questioned. With the gradual decline of the whaling industry, the business passed into less reliable hands, who, tempted by the enormous price offered, could not be restrained from cutting too deeply into the body of the fish. The refinement of fish jaw oil involves a process running through very nearly two years of careful work, so this mixture of inferior oils could not readily be discovered in the crude material.

Some years ago, with the stock of crude material very low, a small batch of oil got upon the market which did not prove up to the mark, and it took some time to discover the cause. Some of this oil was what caused the Navy to abandon its use in gyros. After the discovery of this, the question came up of finding a new and reliable supply, which entailed an enormous expense.

Nearly all the large manufacturers of watches and clocks in this country, knowing fish jaw oil to be the most reliable lubricator for their purpose, gave most kindly interest to the matter. To this end a line of fisheries was established far up and down the Atlantic Coast for the express purpose of catching porpoise, cutting with utmost care, the delicate parts of the jaw from which this oil is made. The only attempt ever made to produce the oil from fish to finish.

Since there is no loss without some gain, in some respects it may be that the failure of the aforementioned small batch of oil proved a blessing in disguise, for it became an incentive for the most careful study of the subject. In the past few years the chemistry of the subject has been most carefully worked out. With extensive fisheries on the Atlantic Coast and improved methods of refinement, it is believed that a thoroughly reliable high grade article, much finer than has even been available before, can now be had in adequate quantities for our needs.

SPECIFICATIONS AND TESTS.

In purchasing a fish oil, such as porpoise jaw, the usual specifications for a mineral oil are not applicable. The animal oils have different characteristics, and each different oil has a set of different values which are peculiar to it, and all of which are peculiar to it alone. It becomes, thus, an easy matter to determine what kind of an oil is offered, and certain of the values will then determine the purity of the oil, and its suitability for any desired use.

The particular set of constants applicable to porpoise jaw oil is as follows:

Specific gravity	.924-.926
Iodine number	22-35
Saponification number	260-295
Reichert-Meisel value (5 grams)	131-139
Viscosity (Saybolt Universal) 100 degrees	95-105
Cold test	not over 10 degrees F.
Refractive index	1.4519 at 20.5 degrees C.
Free fatty acids (as oleic)	1.5 per cent

Of these constants, the two most of interest to us are the iodine number and the free fatty acid (as oleic) content. The acid content needs no explanation; the iodine number is, for all practical purposes, a measure of the drying qualities of an oil, the lower the iodine number the less the oil drys. The iodine numbers given, 22-35 for porpoise jaw oil, are the lowest in the range of animal or fish oils; all others run much higher. It is obvious that these two values are most important in our use, and both must be low. Acid is desirable up to .5 per cent.

The investigation having progressed to the point where it was evident that porpoise jaw oil alone of the pure oils would do our work, a recommendation to that effect was made to the Bureau. The Torpedo Station believed that enough of this oil could probably be obtained, and that if carefully safeguarded the danger from acid was not to be feared. However, the Bureau felt that the cost was far too high (around $100.00 per gallon) and that the supply was too limited and uncertain. Therefore it was directed that research be continued.

DEVELOPMENT OF BLENDS.

There was no way to get around the Bureau's objections. The use of blackfish head oil, an oil of precisely similar chemical composition, but somewhat lighter in body and of less strong odor, was investigated. This oil would do as well as the true porpoise jaw but even so, the amount available was not greatly increased, nor was the cost appreciably decreased. The proposition of blending the oil was therefore again considered. It was known that porpoise oil had been blended with clear mineral oils of the medicinal type, and that the result was reasonably satisfactory in watches, clocks, etc. Indeed, various blends were known and used for various purposes in the lubrication of delicate machinery, the best oils being those with the highest proportion of true porpoise jaw. This in some cases ran up to about 50 per cent. This therefore, seemed a good line of attack.

Accordingly pure porpoise jaw oil was blended 50-50 as a starter with a high grade pure white "Russian" mineral oil. This was tried out successfully in various ways, and the general opinion what that it was quite as good as the pure porpoise jaw, and better than anything else ever tried. In a "breakdown" run of a gyro it stood in the bearings even better than the pure oil. Dr. Gill was again consulted and he stated that the blended oils were probably better than the pure oils, and that each gave to the mixture something that the other lacked. And this appeared to be the case.

A mineral oil will not keep moisture off a metal surface. Moisture appears to get under the oil and push the oil away from the metal. On the other hand the fatty oils, particularly if a small amount of acid (usually expressed as isovaleric) is present, do resist water, and will themselves cling to the metal by a process of adsorption, preventing moisture from reaching the metal surface. The mineral oils, for their part, have much greater penetrating power, and will thus get a film of lubricant evenly distributed on the bearing much better than will the straight fish oil. Also, the fish oils are subject to gumming to a certain extent, pure mineral oils are not so much so. But the mineral oil will suffer evaporation while the fish oil will not.

It thus appears that in a blend the fish oil gives power to cling to the surface of the metal, keeping off water; the mineral oil will

cause a penetrating flow of the oil all over the bearing; the mineral oil will tend to prevent gumming of the fish oil, and the fish oil similarly will reduce evaporation of the mineral oil. Thus each gives to the mixture something the other lacks, and each inhibits in the blend some undesirable quality of the other. Furthermore, it was known that any animal or vegetable oil, when compounded with a mineral oil, increases in acidity at a much slower rate than when not so compounded. This will give, for one thing, much better storage properties than a straight fatty oil will have.

Of course there were other advantages. The supply of mineral oil was ample and thus the supply of the blend was double that of the pure porpoise jaw. Also, the cost of the mineral being small, the cost of the final product was really halved. However, even so, there was not available with certainty more than perhaps twenty or thirty gallons a year of the mixture, and it would still cost nearly $50.00 a gallon. It was certain, though, that we were on the right track, and we went ahead with other blends—to see what else we could find out.

The other fatty oils suitable for blending were found to be neatsfoot, pure winter-bleached sperm, and possibly degras oil or a good acidless tallow oil. The last two were ruled out as we could not be sure of the integrity of supply, that is there was great doubt that we could get the same thing every time. The same was also feared with neatsfoot, but it was known that the Germans had used a neatsfoot, blended or pure, and that they liked it. Dr. Gill also seemed to think it would do. In fact he recommended blending neatsfoot and sperm with any ordinary good light mineral oil, in proportion about 1 fatty to 2 mineral.

Following Dr. Gill's suggestion some neatsfoot oil was obtained and blended with the white mineral oil previously used in the porpoise jaw blend. The same proportions, 50-50, were used. Also, some was mixed with "Vacuoline Extra 'B'," a good light mineral oil in the proportion of 1-2 above. These oils were tried in the gyro testing stand in the Research Department, where the 50-50 mixture proved fairly satisfactory, but the 1-2 mixture was too light and did not stick in the bearings well enough. At the same time some sperm oil was similarly blended and tested. This

oil proved very satisfactory, indeed more so for actual bearing lubrication while running than anything else up to that time. This was somewhat to be expected because of the known "oiliness" of the sperm which gives it great affinity for metal surfaces.

In the gyro testing stand above mentioned we had an ideal method of testing the value of our oils as lubricants. This stand was simply one in which gyros were run at high speed for long periods as more or less of a breakdown test for the bearings. The various oils were tried in turn and at the end of a run the bearings were carefully examined and the condition and remaining quality of the lubricant carefully noted. This did not, however, give us a test for the preservative qualities of our oils. Some equally simple and effective method for determining this important factor was required.

One suggestion of the Engineering Experiment Station at Annapolis, was to take polished steel specimens, well slushed with the various oils, and place them in a 3 per cent salt solution for 24 hours and see which gave the best protection. This was tried with various oils and the results obtained noted, the oils being indicated for each experiment. After the second day's experiment, the neatsfoot and Vacuoline mixture was discarded as unsatisfactory. After three days experiment, it became clear that the straight oils, either mineral or fatty, were not as satisfactory as were the mixed oils. This is believed due to the fact that straight mineral oil lacks power to adhere to metal surfaces when in the presence of moisture or water; that is, water can get underneath the oil and in direct contact with the metal. In the case of the fatty oils, these would not have sufficient body to withstand the action of the salt solution. In the mixed oils, however, the mineral oil furnished the body to withstand the action of the solution while the adhesive qualities of the fatty oil enabled them to stick close to the metal and keep the water from getting under the mineral oil. This is our understanding of the action of the mixed oils and it seems to be borne out by continued experiments.

The result of tests so far put the porpoise jaw mixture best and the other two about the same, the sperm mixture withstanding the corrosion a little longer in the last test, but in the end being a little worse than the neatsfoot mixture. These tests, of course, were

very severe, far more so than the gyro is called upon to withstand in actual service. The fact that these oils stood the tests as well as they did was a pretty good indication that they ought to be satisfactory from the standpoint of preservation of bearing metal.

On 15 December, 1928, Mark VIII-8 torpedo No. 11753, lost on 22 November, was recovered. The gyro was taken out and taken to the Gyro Department for examination. It was found flooded with a mixture of oil, water and mud. The wheel was stiff and turned only with difficulty. It was disassembled and the heavy oil, water and mud cleaned off with gasoline and air. All bearing surfaces were found shiny and clean, no signs of pitting or rust on the balls, ball races or pivots, these parts being just as shiny and bright as the day they were assembled. Some rust was found on the outer steel parts of the assembly which had not been oiled. The oil used in this gyro was the Station porpoise jaw mineral mixture, and it had acted as a perfect preservative in this case. This was a very severe test under a condition that might be encountered any time in service. After the gyro was cleaned, and assembled, it was as good as new. Incidentally, the operatives in the Gyro Department reported this porpoise jaw mineral oil mixture as the best gyro oil they had ever worked with up to this time. The gyros were returning from the range in perfect condition and it gave altogether excellent satisfaction.

At this stage it appeared fairly certain that the solution of the problem lay in a suitable mixture of a pure, heavy, or medium bodied mineral oil with a suitable fatty oil, animal or fish, and of the fatty oils available for mixing, the three mentioned above were believed to be the best, though this was purely a matter of opinion based on general characteristics of these and other fatty oils as given in "Archbutt and Deely" and other authoritative texts, the criteria being low iodine numbers, moderate saponification, and low acid, and ease with which the various oil adulterants may be discovered. In the case of sperm and neatsfoot oils, there can be no objection on the grounds of supply, as unfortunately exists in the case of porpoise jaw oil.

So far, except in one case, the mineral oil used was a pure, heavy, colorless medicinal oil. It appeared not unreasonable to suppose that an ordinary high grade light commercial lubricating

oil might serve equally well as the medicinal oil. It was realized that certain specifications must be set that would probably be more severe than the usual Navy Department specification. It was clearly important to have little or no carbon residue, and no gumming; and desirable to have a very clear and light colored oil. In meeting these requirements the Engineering Experiment Station again came to our rescue and recommended a trial of the Gulf Refining Company's "Gulfpride" oils of suitable viscosity.

In January, 1929, we received from the Gulf Refining Company three gallon samples of these oils: light, extra light and H.S., with viscosities at 100 degrees F. of 200, 150 and 100 seconds Saybolt. These oils were described as a very high grade of light lubricating oils, non-gumming and non-drying. They were desired for test mixed with fatty oils as a possible substitute for the white Russian oil previously used. Also, about the same time, samples of the highest grade sperm oil and neatsfoot oil were obtained from the Wm. F. Nye Company, these oils being much purer than the ordinary commercial qualities such as is obtained on Navy contracts. A gumming test was performed on the Gulfpride oils and also with the white Russian oil previously used, both of which showed negative. No tendency to gum should therefore be expected in either of these oils.

In order to test the behavior of the Gulfpride oils when mixed with fatty oils in comparison with the white Russian mixtures, mixtures were made up as follows: one-half porpoise jaw and Gulfpride light, sperm oil and Gulfpride light, and neatsfoot oil and Gulfpride light. Samples of the pure sperm and neatsfoot oils were mixed with white Russian oil, half and half. Samples of the previous mixtures with white Russian oils were also taken. Steel test pieces were slushed with these oils and immersed in a 3 per cent salt solution for varying periods and the actions closely watched. The porpoise jaw and Gulfpride mixture in each case was the best, its performance as a rust preventive being most remarkable. A steel sample remaining in the solution from Friday morning until Monday morning showed practically no evidence of corrosion at the end of this time. The steel pieces immersed in the other oils were all more or less corroded.

These tests were repeated over a period of one week. The comparative grading of their rust preventive powers would be first, porpoise jaw and Gulfpride; second, neatsfoot and Gulfpride and porpoise and Russian about the same; and fourth, sperm and Gulfpride nearly as good as the preceding two. On some occasions, there would be slight variations in the order in which the second, third and fourth would follow but the above is a close average.

The viscosities of these mixtures at 100 degrees F. were: Porpoise jaw and Gulfpride, 126 seconds Saybolt; Neatsfoot and Gulfpride, 194 seconds; Porpoise jaw and white Russian, 135 seconds; Sperm and Gulfpride, 125 seconds. The viscosity of the neatsfoot and Gulfpride is too heavy and that mixture was definitely not suitable for gyro use for that reason. In order to test a neatsfoot mixture of proper viscosity, a mixture was made up with neatsfoot and Gulfpride H.S. (100 seconds) and was tried out in the salt solution. This mixture gave a suitable viscosity, but was entirely unsatisfactory as a rust preventive. This fact, together with the other considerations to be mentioned later, definitely removed neatsfoot oil from consideration for our purposes.

As a further test with the object of ascertaining the best percentages in which to mix the fatty oil with the mineral oil, a mixture of porpoise jaw and Gulfpride Extra Light (150 sec.) was made in a proportion of 1 part porpoise to 2 parts Gulfpride. This gave about the desired viscosity, 125 seconds. Steel samples slushed with this mixture were put in the salt solution and given the same tests as previously. These samples showed considerable corrosion in a few hours and were covered with rust in 24 hours. The Gulfpride oils were used alone for a similar test in order to see what the action was. Steel pieces slushed with these oils were quite badly corroded in 24 hours. As a result of these experiments, it now appeared that as a preservative the Gulfpride and porpoise jaw 50-50 solution was the best we had, closely followed by the half sperm and Gulfpride.

In order to test out the lubricating qualities of the two mixtures, samples of them were sent to the experimental gyro stand to be used in lubricating gyro running tests at high speed and for long

periods. Heretofore the old Station mixture of half and half porpoise jaw and white Russian had been considered the best of the oils but the porpoise and Gulfpride now proved pronouncedly superior in adhesive qualities and as a lubricant generally. A great deal more oil remained in the wheel bearing than with any other oil yet tried and it appeared to afford a better lubricating film in them. The sperm mixture gave results equally as good.

The second place in our fatty oil list was close between pure neatsfoot oil and pure sperm oil. The final decision was made in favor of sperm oil for reason as follows: It is well known that the term neatsfoot oil can mean any one of a number of oils pressed from the hooves and bones of various animals. In fact, in some cases sheep grease is pressed and then sold as neatsfoot oil. It is thus very difficult to obtain a standard quality of this oil because it is impossible to tell from the oil what it comes from, and unless you get oil consistently from the same sources you will not get the same qualities in the oil. It is not as easy to get pure oil in this class as it is in the case of sperm. You can order the highest grade of neatsfoot oil but never can be certain of what you get. Also, the viscosity is too high and when mixed with a light enough oil to reduce the viscosity, the mixture may not have enough protective power. For these reasons, further consideration of neatsfoot oil was abandoned.

In regard to the sperm oil, Dr. A. H. Gill writes: "In my opinion, a mixture of sperm oil with a mineral oil of lower viscosity than Russian you mentioned would be ideal for the purpose. The only difficulty feared in the case of sperm oil is its tendency to gum or to oxidize, which is higher in the case of sperm oil than with the porpoise jaw oil."

Tests were continued with the two most satisfactory mixtures, running them in the high speed gyro tests to develop their lubricating qualities, which proved excellent in both cases. The oils were also put through oxidation and evaporation tests and the standard tests for gumming. No gumming was found and no appreciable evaporation or thickening under the oxidation test. It was interesting to note the immensely superior performance in every respect of the blended oils over either of their constituents in the pure state. It seems certain that each oil in the blend sup-

plies some quality to the mixture which the other lacks and also inhibits some undesirable quality which the other possesses.

Continuing the search for oils suitable for blending, the Station obtained from the William F. Nye Company, of New Bedford, Mass., three samples of blackfish oil produced by them. No. 1 was oil obtained from a school of blackfish that went aground on Cape Cod in the summer of 1928 and was wholly refined under the direct supervision of the company's officers at their main plant. No. 2 was from fish caught off the Labrador coast and No. 3 was from the Arctic fisheries. Nos. 2 and 3 had a considerable part of the refining done on the spot and evidently not under as close supervision all the way through as No. 1. The company stated that the tests on these oils would probably rank them in the order indicated. These oils were blended in the usual manner and put through the usual tests, here, and the results bore out the prognostications of the company. No. 1 oil was practically as good as the true porpoise jaw oil. Nos. 2 and 3 were not satisfactory.

One thing that was shown by our experiments, and particularly those on blackfish and sperm oils, is the necessity for a very careful and high refinement in these oils. This is a very difficult matter to express in terms of specifications. It probably could only be done by means of an acceptable list on which would appear companies whose product had been proven satisfactory. In the case of sperm oil, the specifications would have to include a cold test, although it is not yet certain exactly how low it can be brought, probably somewhere between 25 degrees and 30 degrees F. The present Navy specifications do not set any cold test. This is a very important indicator of the refinement of the oil. In any case, the Station believes that it has in the corrosion test, as at present conducted, a fairly good means of ascertaining whether the oils received have been carefully enough refined and are satisfactory for our use.

At this stage a curious reversal of findings took place, sperm mixtures displacing porpoise jaw in a measure as the best preservatives. Hitherto ordinary Navy specification sperm had been used. With the specially refined article now obtained, results changed.

The reason for this reversal of our previous experience was sought in various directions but so far no satisfactory explanation has been found. It was thought that perhaps the oils were warm when put on the steel and so, to see whether the temperature of the oil made any difference, some oil was put on when cold and the specimens were left outside. These specimens outside showed more than twice the length of time before corrosion set in than did those inside. Here, again, the sperm mixtures stood up better than the porpoise jaw.

With the idea that perhaps the age of the porpoise jaw used in these tests, although it was approximately the same as the age of the oil previously used successfully, might have something to do with the reversal of standing in these tests, two quarts of new porpoise jaw oil were obtained from the Wm. F. Nye Company. The oil obtained, however, turned out to be blackfish head oil of the same lot as that previously tested on the Station here as blackfish No. 1. The chemical characteristics were well within the specifications for Station porpoise jaw, the acid content very much lower in fact. This oil was mixed up and samples put inside and outside together with samples of some sperm mixtures. These blackfish oil mixtures gave about the same results that the porpoise jaw mixtures had previously given, perhaps a little better, but still not as good as the sperm mixtures.

The sperm oil used in the first tests described was from Station sperm, the small sample of highly refined sperm having been used up. The second tests were conducted with the highly refined sperm, a further supply having been obtained from the Wm. F. Nye Company. The highly refined sperm, as was to be expected, stood up better than the ordinary Station sperm. In all these tests, those carried out on the outside lasted longer than those placed inside. This gives evidence that the oils when issued should be kept in a cool place and suggests the advisability of keeping the gyros themselves, when laid up, in a cool place. These tests also indicated that the properly refined blackfish oil will be quite as satisfactory in the mixtures as true porpoise jaw and that the specifications as laid down to include blackfish oils are correct.

An examination of the gyro bearings oiled and laid up in the constant temperature oven in the chemical laboratory and another

set oiled and laid up under outside atmospheric conditions was made. These bearings had been laid up from three to six months. The oils usd were the various porpoise jaw, sperm and blackfish mixtures. The bearings in all cases were entirely free from any corrosion or any serious gumming. In the case of one bearing oiled with one of the sperm oil mixtures where apparently an excessive amount of the oil had been put in, there appeared to be thickening of the oil between the balls in the cup, but the balls themselves moved around and rotated on each other perfectly freely. In the samples of oils in open glass cups in the constant temperature oven, there was no apparent oxidation or gumming or thickening of the oils except possibly in the case of sperm. This thickening or gumming of the sperm is the greatest drawback to the use of this oil in the mixtures but both Dr. Gill and the Bureau of standards appear to believe that by properly refining the sperm, its tendency to gum can be so reduced as to be negligible.

With the satisfactory development of the two oils, and the fixing of their proportions and characteristics, it was felt that the problem was well on the way toward solution. However, difficulties still remained. When fresh the two oils were fine, probably equally good. But the porpoise jaw mixture was still expensive and severely limited in available quantity, while the sperm mixture was always open to the suspicions of gumming.

THE ANTIOXIDANT.

At this juncture the Bureau of Standards, which had been consulted, reported that the R. T. Vanderbilt Company, of New York had done some work with antioxidants, substances to be added to other substances to prevent absorption of oxygen with consequent changing of characteristics. They had developed their substances primarily in connection with the rubber industry, but had to some extent tried them in oils.

It was at once realized that if we could prevent the absorption of oxygen by our oils, we would have very nearly solved our problems. The changes to be feared and guarded against in these oils namely: thickening, gumming, increase in acidity, becoming rancid, were due almost entirely to oxidation. Therefore, if we

could prevent the increase of oxygen in the oil, we would very nearly have fixed the characteristics of the oils for good and all. With this end in view the Vanderbilt Company was approached.

They were found most willing to collaborate with us. Their chemists thought that they had an antioxidant, a preparation of adol-alpha-napthylamine * which would do the business. This preparation acts as a negative catalyst; that is, while effecting no change in the chemical properties or characteristics of the oil, it prevents the absorption of oxygen by the oil with consequent increase in acidity, rancidity, gumming, etc. It itself has no effect whatever on the oil's lubricating power, and only a small, and that good, effect on its preservative qualities.

In order to make a test of the possibilities of their substance, a quantity of our two oils was sent to the Vanderbilt Company's laboratories, and a personal visit made there to witness the tests. The method was as follows: A quantity—.2 per cent of the antioxidant by weight was added to the oils, and incorporated in them by warming until thoroughly dissolved. The oils were then put in flasks of pure oxygen, and kept there at a constant temperature of 80 degrees C. for 72 hours. From time to time the amount of oxygen absorbed was measured and at the end the change of acidity was measured. The results, showing oxygen absorbed by treated and untreated oils is given in the following table:

Oil Sample	Oxygen Absorbed (C.C. Per 100 Grams)	Acidity (Initial) Per Cent	Acidity (Final) Per Cent	Increase Per Cent
Sperm, Untreated	1460	0.24	2.66	2.22
Sperm, Treated	408	0.24	1.74	1.50
Porpoise Jaw, Untreated	820	0.40	3.10	2.70
Porpoise Jaw, Treated	83	0.40	0.41	0.01

The results were so startling as to create great interest and enthusiasm in everyone concerned in the work. It certainly looked as though the work was to be crowned with success.

A quantity of the antioxidant was obtained from the Vanderbilt Company and incorporated in the oils on the Station. These oils were then used on the steel test-pieces in comparison with untreated oils, and in every case showed better preservative powers. Appar-

* Patented.

ently the antioxidant gives them superior oxygen resisting powers, indeed this was to have been expected. On tests in gyros in service and on the stands the same result was noticed, and the lubricating qualities were seen to be unchanged. The operatives in the Gyro Department were questioned at length, and each man individually was found to prefer the new oil he was using to any other he had ever seen. Some were using the treated porpoise jaw blend, and some the treated sperm blend. Careful watch could detect no appreciable difference in the performance of these two blends.

After a while, however, a curious sort of colloidal precipitate was noted in the treated oils. This was a source of worry, as such a condition would be very serious. Filtration was tried with apparent success, and lest, by filtration we had removed some of the antioxidant, we sent unfiltered and filtered samples to the Vanderbilt Company for test. The results, taken with extreme care are given below:

Oil	Oxygen Absorbed (C.C. Per 100 Grams)	Acidity (Initial) Per Cent	Acidity (Final) Per Cent	Increase Per Cent
Sperm, Unfiltered	55	0.20	0.31	0.11
Sperm, Filtered	40	0.20	0.24	0.04
Porpoise Jaw, Unfiltered	52	0.62	0.64	0.02
Porpoise Jaw, Filtered	37	0.62	0.62	Nil.

A very remarkable series. Obviously filtering had no bad effect on the qualities of the oils.

However, it soon appeared that this process of filtration, an extremely slow one, had not solved the problem. The previously filtered and clear oils again developed a precipitate. After a long series of tests it was finally found that this was caused by too high heat in melting the powdered antioxidant into the oil. If a heat over 200 degrees F. was used, precipitate formed in time, if the temperature was held between 150 degrees to 200 degrees F., the oil remained perfectly clear indefinitely.

The next thing was to watch performance closely, and this was done all through the summer range work. Nothing but praise was heard for the new oils; they were indeed found excellent not only

for the gyros, but for all other light mechanisms in the torpedo. Not only that, but the oil appeared excellent for all sorts of similar gear, gyro compasses and steering gears, fire-control instruments, fishing reels, guns, camera shutters—anything that needed very occasional but very careful lubrication.

Finally it appeared that the two oils would be equally satisfactory and stable. As the cost differential and the quantity available is infinitely in favor of the sperm mixture; that blend was recommended for adoption for the purposes of the Bureau of Ordnance. The porpoise jaw blend, however, still remains as the most interesting one, and indeed its use in watches, chronometers and other extremely fine mechanisms, is to be preferred of the two. If considerable space in this article has been given to porpoise jaw oil it is because of its very interesting source, production, characteristics and value, and also due to the fact that for so long a time it occupied the major place in our investigations.

At any rate, here is something new and valuable in oils, something that all who are concerned in the work feel will be of increasing commercial importance. And once more the *Navy* has lead the way.

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It is not too much to say that without their help the problem could not have been solved.

DYNAMIC BALANCE OF MACHINERY.

By Lieut. F. A. Tusler, (CC) U. S. Navy, Member.

The purpose of this article will be to cover, first, the theoretical aspects of dynamic unbalance, and second, to describe various types of balancing machines on the market and the underlying principles of each one.

UNBALANCE AND HOW CORRECTED.

Before discussing the general problem it may be well to review one law of rigid dynamics. The effect of rotating a mass is to produce a centrifugal force which acts on the shaft to which attached. In Figure 1 assuming a rigid weightless shaft, let:

ω = Angular speed in radians per second

r = Distance of center of gravity of mass from center of shaft

$M = \frac{W}{g}$ = mass under discussion

F = centrifugal force due to the rotation of M.

$$\text{Then, } F = M.\ r.\ \omega^2 = \frac{W}{g}.\ r.\ \omega^2 \quad . \quad . \quad (1)$$

If the rotation is uniform, as can be assumed for this work, the line of action of F passes through the axis of the shaft. It is readily seen that such a force continually changing in direction will cause the shaft to exert forces on its bearings, the effect of which is vibration. If the unbalanced mass lies in a transverse plane through the center of gravity, the axis of the shaft will tend to generate the surface of a right circular cylinder, as illustrated in Figure 2.

Consider next Figure 3, which represents two equal masses at equal distances from the axis of the shaft, lying on opposite sides of the shaft and separated from each other axially by the distance

d. Using the same notation, we have, when the shaft is rotated at the speed ω, two forces acting as shown which form a couple whose magnitude may be expressed as:

$$F.\ d = M.\ r.\ \omega^2.\ d = \frac{W}{g}.\ r.\ \omega^2.\ d \quad . \quad . \quad . \quad (2)$$

Again forces are exerted on the bearings of the shaft due to the action of the couple and the shaft will tend to generate the surface of a double cone as shown in Figure 4.

STATIC UNBALANCE.

Following the customary nomenclature, a rotor is considered in *static unbalance* when the conditions illustrated by Figure 5 obtain. Here we have a rotor, all of whose mass is symmetrically located with the exception of W, which mass is assumed to lie in a transverse plane through the center of gravity of the rotor at a distance r from the axis. In this case of course, the center of gravity does not lie in the axis. It is evident that the amount of such unbalance could be determined approximately by carefully rolling the shaft on hardened knife edges, and that a fairly satisfactory balance could be obtained by adding a correction weight in the transverse plane through the center of gravity directly opposite the unbalanced mass of amount, such that its moment about the axis is equal to the moment of the unbalanced mass, or equal to W.r.

DYNAMIC UNBALANCE.

By *dynamic unbalance* is meant a condition such as shown in Figure 6. In this case the rotor is in perfect static balance, but, if rotated, forces arise of the same character as those shown in Figure 3. Obviously the amount and location of this unbalance cannot be determined by rolling the shaft on knife edges.

Experience has shown that, in general, rotors are both statically and dynamically out of balance, but, as the separation of the unbalance into its static and dynamic components serves no useful purpose, this resolution will not be explained. The combined static and dynamic unbalance is what must be corrected, and a method for its determination follows.

Actually the unbalance of a rotor is due to certain masses which are unsymmetrically located with respect to one another. When the rotor is rotated, each of these masses gives rise to a centrifugal force, the magnitude of which is determined from Equation (1). If one bearing, say the right end, of an unbalanced rotor is fixed with respect to vibration and all centrifugal forces due to the above mentioned unbalanced weights are referred to a transverse *reference* plane, say to the left of the fixed bearing end, it can be proved that, by resolving all such centrifugal forces into forces and couples and transferring them to the reference plane, the resultant in the reference plane is a *single* force; (a second force of different magnitude and acting in the same direction will lie in the transverse plane through the fixed bearing, but as we are dealing only with forces about the fixed bearing, this second force does not concern us). The moment of the resultant force in the *reference* plane about the fixed end is equal to the vectorial sum of the moments of the individual forces about the fixed end. This resultant force may be considered as a "heavy spot," *i.e.*, as a certain weight, in a definite axial plane, at a certain radius from the axis, and at a certain distance from the fixed end. If now an equal weight is placed at the same distance from the axis in the reference plane, but on the opposite side of the axis from the "heavy spot," the rotor can be rotated at any speed without vibration of the unfixed end, but there would still exist an unbalanced force at the fixed bearing. All balancing machines and instruments for determining unbalance make use of this fact. One determines the force in the *reference* plane by cancelling it with an equal and opposite force by means of an adjustable balance weight; one machine by observing the amplitude of vibration of the free end by means of a calibrated spring; one measures the force directly by means of a calibrated spring; one by measuring the force in a manner similar to the method of the beam balance; and another by measuring the amplitude of vibration set up in the bearing of the machine itself.

The product of the balance weight opposite the "heavy spot" by its radial distance from the axis will be designated by the symbol R if the right end is considered fixed; by R′ if the left end is fixed. Strictly speaking, we are dealing with centrifugal forces, but, since all elements of the rotor rotate at the same angular

speed, these centrifugal forces are always in the same ratio and proportional the product of the weight by its distance from the axis, so that the centrifugal force may be more simply regarded as the above product, leaving out all constants.

Since correction weights will in general be added in some transverse plane other than the reference plane, it becomes necessary to transfer the above resultant force R (or R′) in the reference plane to the *correction* plane. In Figure 7, let A be a force in correction plane A, whose moment about the fixed end is equal to the moment of R about this end.

$$A\,(1 + f) = R.\ a$$

$$\therefore A = \frac{R.\ a}{1 + f} \quad . \quad . \quad . \quad (3)$$

In adding correction weights to our rotor to cancel the force A, care must be taken that the centrifugal forces of these correction weights have the same moment about the fixed bearing as force A, and that the moment of these forces about the free bearing is zero. The second condition is necessary, since the A connection weights will not be in place when the B end of the rotor is balanced and were this condition not fulfilled, the addition of the A correction weights would upset the balance of the B end. Obviously we must add weights in both correction planes to meet such a requirement. Let A_A be a force in correction plane A, which, in conjunction with a force A_B in the correction plane B, will have the same moment as force A about bearing B, and which forces taken together have no moment about bearing A. It is required to find A_A and A_B. Taking moments about bearing A.

$$A\,(1 + f) = A_A\,(1 + f) - A_B\,.\,f \quad . \quad . \quad (4)$$

Taking moments about bearing A:

$$A_A\,.\,e = A_B\,(e + 1) \quad . \quad . \quad . \quad (5)$$

Substituting in (4) the value of A from (3), we get:

$$\frac{R\ a\ (1 + f)}{(1 + f)} = A_A\,(1 + f) - f.\ A_B$$

$$A_A\,(1 + f) = R.\ a + A_B. \quad . \quad . \quad (6)$$

Eliminating A_A from (5) and (6), we get:

$$A_B = \frac{R.a.e}{L.l} \quad . \quad . \quad . \quad (7)$$

Substituting in (5) the value of A_B from (7):

$$A_A = \frac{R.a\ (e+l)}{L.l} = \frac{R.a}{L}\left(1 + \frac{c}{l}\right) \quad . \quad (8)$$

It should be noted that the force A_A lies in the correction plane A, and in the axial plane containing R, and on the same side of the axis; that force A_B lies in the correction plane B and the axial plane containing R, but on the *opposite* side of the axis.

Considering now force R′ that was obtained by holding the A end fixed and allowing the B end to vibrate, we obtain two forces, one:

$$B_A = \frac{R'.b.f}{L.l} \quad . \quad . \quad . \quad (9)$$

which lies in the correction plane B and in the axial plane containing R′ and on the same side of the axis as R′; and:

$$B_B = \frac{R'.b}{L}\left(1 + \frac{f}{l}\right) \quad . \quad . \quad (10)$$

which lies in the correction plane A and the axial plane containing R′, but on the opposite side of the axis.

Forces A_A and B_A in the A correction plane can now be composed graphically as shown in Figure 8, yielding the resultant R_A; similarly, the forces B_B and A_B in the B correction combine to form resultant R_B. If R and R′ were obtained in terms of inch-ounces as is usual, the correction weights to be added in the correction planes A and B can be calculated as soon as the radius at which it is desired to place these weights has been settled, by dividing R_A and R_B by their respective radii. If r_A and r_B be these distances, the A correction weight is:

$$W_A = \frac{R_A}{r_A} \text{ (ounces)} \quad . \quad . \quad . \quad (11)$$

The B correction weight is:

$$W_B = \frac{R_B}{r_B} \text{ (ounces)} \quad . \quad . \quad . \quad (12)$$

While this may sound rather formidable, the actual calculation is quite simple. To recapitulate: the balancing machine gives a measure of the unbalance in the reference plane in inch-ounces for each end separately, denoted by R and R′; A_A, A_B, B_B, and B_A are computed by formulas (7), (8), (9), and (10), which work can be simplified if arranged in the form illustrated in Figure 14(a); the vector diagram of these forces is laid out as shown in Figure 15, and the correction forces obtained graphically; the radius at which the correction weights are to be attached is usually fixed by the form of the rotor, so that the calculation of the weights proper involves only the division of forces R_A and R_B by their corresponding radii.

It sometimes happens that the plane in which it is desired to add the correction weights is located outside the bearing, *e.g.*, in certain turbo-blowers when balanced with blower attached to the turbine rotor and in certain generators which are provided with balance holes in flanges coming outside the bearings, as illustrated in Figure 9. In this case it will be noted that the distance e is *negative* and cognizance must be taken of this negative value in calculating A_A and A_B. A_B will be negative in sign and will therefore be plotted in the direction opposite from that shown in Figure 8.

In locating a correction weight it often happens that this weight cannot be located in the axial plane of the forces R_A or R_B, in which case it is convenient to split the correction weight into two parts. In order that the two new weights will have the same effect as the former single weight, the construction shown in Figure 10 is resorted to where R_B is replaced by R'_B and R''_B.

The following procedure may be used when the counterbalance furnished with the balancing machine is insufficient to bring the rotor in balance. Certain main motor armatures for the R-type submarines were so badly out of balance that an auxiliary balance weight had to be used in addition to the regular balance clamp. The manner in which the auxiliary balance weight is combined with the regular balance weight follows:

In Figure 11, let:

W = weight of the regular balance weight.

r = distance of the balance weight from the axis.

N = weight of the auxiliary weight.
v = distance of the auxiliary weight from the axis.
W.r = Rc = moment of balance clamp about the axis.

First, transfer the moment N.v from the plane through M to that through P, containing the balance clamp. Taking moments about B,

$$(1 + f)N.v = R_A.a$$

where R_A is the moment the auxiliary would have, had it been in the reference plane through P

$$R_A = \frac{N.\ v\ (1 + f)}{a}$$

The total unbalance R is now obtained by composing Rc and R_A graphically as shown and lies in the axial plane containing R.

The establishment of a criterion by which engineers could judge whether or not a certain type of machine requires a dynamic balance would tend to considerable savings by the elimination of balancing work now undertaken where practical considerations would seem to indicate none necessary. It is not the purpose of this article to set up such a standard, but an attempt will be made to indicate how the problem may be approached.

Each particular type of rotor has a characteristic range of unbalance within which individual rotors of the type fall. This range is soon established after a few rotors have been balanced. If now the force acting at the bearings due to unbalance is known, the engineer is in a position to decide whether or not it is likely to cause bearing trouble or undesirable vibration effects on the foundation. In this connection it must be remembered that, if the speed of the rotor coincides with the natural frequency of the bearing pedestals or foundation, a small amount of unbalance in the rotor will build up a large amplitude of vibration. Fortunately such phenomena are rarely found in practice.

Reference to Equation (1) shows that the effect of rotating a mass M at a distance r from the axis at ω radians per second is a centrifugal force of amount, $M.r.\ \omega^2$ pounds, where $M = \frac{W}{g}$.

Consider now the results obtained by the balancing machine. Referring to Figure 14(b), the unbalance is either expressed in inch-

ounce-inches, which has the following significance: the unbalance in the rotor with the B end fixed has been eliminated at the A end by the addition of a weight in the reference plane through P of amount, W ounces, out a distance, r inches, from the axis, and at a distance, a inches, from the B bearing; or in inch-ounces, in which case the distance of the balance clamp from the fixed bearing should be specified. If we take as our reference plane, a transverse plane through the bearing A, our unbalance would be W′.r′.L, such that W′.r′.L′ = W.r.a. Let the force acting at the A bearing be F_A

$$F_A = \frac{W.r.a.\omega^2}{g.L} \text{ (pounds)} \quad . \quad (13)$$

If the angular speed is in R.P.M., this becomes:

$$F_A = \frac{W.r.a}{32.2\,L}\left(\frac{2.\pi.n}{60}\right)^2 = 3.41\left(\frac{N}{100}\right)^2\frac{W.r.a}{L} \quad (14)$$

Where,

N = R.P.M.

W = weight in pounds.

a and L are distances in feet.

If a and L are in inches and W in ounces, the force F_A in pounds is given by:

$$F_A = 1.774\left(\frac{N}{1000}\right)^2\frac{W.r.a}{L} \quad . \quad . \quad (15)$$

TYPES OF BALANCING MACHINES AND PRINCIPLES UPON WHICH BASED.

THE AKIMOFF BALANCING ROLLS.

In Figure 17 the journal J of the rotor is free to rotate on the rolls R, running in self-aligning ball bearings, which in turn are supported by the vibrating frame V. This frame is supported from the stationary frame F by a thin steel section K, which acts as a virtual knife edge for the vibrating frame V and which allows the journal J to oscillate about K, but which oscillation may be considered as a transverse vibration in a horizontal plane for the journal, providing the range of oscillation is limited to small angles. The helical springs S are introduced between the vibrating and

stationary frames to vary the natural period of the vibratory system and limit the amplitude of vibration. The adjustable weights W, carried on cantilever arms, are used to vary the natural period of vibration. Jacks L provide a means of locking the vibratory system in a stationary position and act as limit stops when the system vibrates. A friction brake (not shown) between the vibrating and the stationary frame can be used to dampen excessive vibration.

One of these rolls is used at each end of the rotor to eliminate shifting the rotor end for end. A balance clamp is furnished, together with a number of split bushings so that the clamp may fit shafts of various diameters. This clamp is in perfect balance when the weights are in the position shown in Figure 18, since one weight is double the other. Figure 19 shows the arrangement of the rolls for balancing a destroyer propeller.

Rotation of an unbalanced rotor sets up forces proportional to the square of the speed. With reference to any horizontal axial plane these forces have a period equal to that of the rotating body and vary in intensity in this plane during one revolution as simple harmonic motion. Advantage is taken of the fact that, if an oscillating system is acted upon by a system of external periodically varying forces, the period of which equals the natural period of the oscillating system, a resonance effect is produced, *i.e.*, the amplitude of vibration is greatly increased over that produced when these periods are out of phase with one another. By designing the oscillating system with a low natural period of frequency, the speed at which the rotor must be rotated to set up this resonant vibration is held to a low figure. From Equations (1) and (2) it will be noted that, at the same speed, the forces due to unbalance are directly proportional to the amount of unbalance, so that it is fairly easy to determine whether or not the balance has been improved by the shift of a balance weight.

The rotor is placed on the balancing rolls as shown in Figure 14(b) with one end, say the "B" end fixed, and the other, the "A" end, free to oscillate in a horizontal plane. The rotor is belt driven from directly below to eliminate vibration from this source. At the resonant speed the amplitude of vibration of the vibratory system is noted and on subsequent runs the balance clamp adjusted

as to angular position and the balance weights run out from the axis by trial and error until the balance clamp neutralizes all unbalanced forces acting about "B." Similar data is obtained by locking the "A" end and allowing the "B" end to vibrate. The balance clamps are considered located in the reference planes through P and P′, so that the balance weight multiplied by the distance it has been moved from its zero position, say for the "A" end free, is identical with the R used in Equations (7) and (8), and this value can be substituted in these equations for the determination of correction weights A_A and A_B; similarly for the "B" end. For the graphical composition of A_A with B_A, and of B_B with A_B, it is convenient to lay off the planes of the clamps as viewed from the "A" end. Figures 14(a) and 15 show such a problem worked out.

The chief advantage of this machine is that it will accommodate rotors of practically any form or dimensions, since the rolls are portable, and the parallels upon which mounted can be adjusted as to location and height to suit conditions. As most Navy Yard work is of a varied nature, these machines are practically the only ones used. The greatest disadvantage is the necessity of stopping the rotor to adjust the balance clamp in the trial and error method of eliminating unbalance. If a suitable clamp could be devised which could be adjusted with the rotor in motion, the time required for balancing would be materially reduced. The calculation of correction weights is also somewhat more laborious than for some of the machines about to be described.

THE WESTINGHOUSE METHOD OF BALANCING ROTORS IN THE FIELD.

Examination of Equations (1) and (2) shows that, at any *particular speed* of rotation, the forces and couples arising from unbalanced masses are directly proportional to the product of the masses and their distance from the axis, *i.e.*, the amplitude of vibration set up is proportional to the unbalance. All balancing machines which obtain the unbalance directly from one or more runs use this fact, although the application is not always evident at first sight.

The Westinghouse Electric and Mfg. Company has devised a method for balancing rotors in the field while running under their own power at normal operating speed. Since this eliminates the necessity of lifting or removing rotors from their bearings,

together with the attendant clearing away of pipe lines, portable deck plates, etc., which incidental work in the case of main propelling units is costly, the possible savings run into big figures. The scheme seems particularly well adapted to marine work.

A detailed description of this method of dynamic balancing was presented in the February 7, 1928, number of "Power" by Mr. G. B. Karelitz of the Research Department of the Westinghouse Company. In the following description, only the general features of the method will be outlined.

From Figure 7 and the attendant description it was found that, if one end of the rotor, say the "B" end, were fixed and the other end, say the "A" end, free to vibrate, all the forces and couples of unbalanced masses could be considered as concentrated in a single resultant force R in the reference plane. This force R may be looked upon as a "heavy spot" in this flange as shown in Figure 20.

It is assumed as mentioned above, and practice approximately confirms the fact, that the amplitude of vibration in a bearing pedestal due to unbalance at a *particular* speed is directly proportional to the amount of unbalance. A sensitive instrument measures the amplitude of vibration of the bearing pedestal very accurately at the high frequency met when operating at the normal running speed.

In Figure 21 (a), 0 is the axis of the rotor and D the "heavy spot" referred to above. At a particular speed, this "heavy spot" will set up a vibration of the bearing pedestal of amount 00′, this vibration, of course, being caused by the centrifugal force R acting in the direction 00′. Suppose now that the rotor is in *perfect balance* and that a trial balance weight of known amount is added in position A, Figure 21 (b). Running the rotor at the *same* speed as before, this trial weight will set up a vibration whose amplitude is 01, and its centrifugal force will act in the direction 01. The *same* weight in position B will set up a vibration of equal amount, 02, but the direction of its centrifugal force is 02. Similarly, the same weight in position C sets up a vibration of amplitude 03, the centrifugal force acting in the direction 03.

Return now to the *unbalanced* rotor with the "heavy spot" D and let the trial weight be placed in position A. The amplitude of

vibration is now the resultant of amplitudes 00′ and 01. The resultant is 01′, acting in the direction 01′. By placing the trial weight in position B, the resultant is 02′, acting in direction 02′. Similarly, we get an amplitude of vibration 03′ by placing the trial balance weight in position C, the resultant centrifugal force acting along 03′. From the construction arrived at in Figure 21:

0′1′ is equal to and parallel to 01.
0′2′ is equal to and parallel to 02.
0′3′ is equal to and parallel to 03.

Therefore, 0′ is the center of a circle whose radius is equal to 01, 02, and 03; also equal to 0′3′, 0′2′, and 0′3′.

Suppose now we have a rotor whose unbalance is unknown. By considering one end at a time and making all runs at the *same speed* dial indicator readings are taken of the vibrations of the bearing pedestal.

Let the first run be made without the trial balance weight attached. The amplitude of vibration will be 01′. During this run chalk mark the "high spot" on the shaft. The "heavy spot" will be approximately 90 degrees ahead of this chalk mark. As shown in Figure 22, mark off three positions on the opposite side of the shaft from the approximate position of the "heavy spot" and place the trial balance weight in position A. Rotate the rotor at the same speed as before and record the amplitude of vibration. This will be 01′ of Figure 21. Stop the rotor and shift the balance weight to position 2 and rotate again, obtaining this time 02′ of Figure 21. Similarly for the trial weight in position C we get 03′ of Figure 21. On a polar diagram, Figure 23, ruled with concentric circles, lay off the angular positions of A, B, & C. From Figure 21, we know that the end 0′ of 00′ falls on the center of this diagram; one end of 01′ along 0′A; one end of 02′ along 0′B; one end of 03′ along 0′C. The other ends of these four lines meet at the point 0. On a four arm protractor mark off the lengths corresponding to 00′, 01′, 02′, and 03′. Keep the mark 0′ of the leg 00′ on the center of the diagram and slide the other three marks along their respective radii until all three marks lie on the *same* concentric circle, dividing any small error between them. Figure 23 shows the protractor as finally arranged.

In Figure 21 note that the radius $0'1' = 0'2' = 0'3'$ corresponds to the amplitude of vibration due to the trial balance weight alone with the rotor in perfect balance otherwise, and that $00'$ corresponds to the amplitude of vibration due to the unbalance only.

$$\text{Therefore, } \frac{\text{Amount of Unbalance}}{\text{Trial Weight} \times \text{Distance from Axis}} = \frac{00'}{0'1'} \quad (15)$$

$$\text{or Amt. of Unbal.} = \frac{00'}{0'1'} \times \text{Wt.} \times \text{Dist. from the Axis.}$$

The amount of unbalance from equation (15) is the R of equation (7) and equation (8) and the correction weights A_A and A_B can now be calculated. Note also that 0'0 is the plane in which a correction weight could be placed that would eliminate all unbalance at the A end. Repeating the above procedure for the "B" end gives R' for Equations (9) and (10).

If a balance clamp has been used in place of trial weights, the result of Equation (15) can be set on the clamp, and its angular position adjusted to that of 0'0. The A end should now run without vibration. If a small amount of vibration is still present, the balance clamp can be left set, and the four runs repeated using small trial weights attached to the clamp. The results of the second series of runs are worked up as before, preferably on the same diagram so that the two results can be combined graphically to give the final R or R' as the case may be.

It is not believed that the result obtained by using this scheme has the same accuracy that is possible with the Akimoff machine, but, since the unbalance left uncorrected would produce vibrations of an amplitude two small for detection in the bearing pedestal, the result for all practical purposes should be satisfactory. The balance of a rotor at its normal running speed is important due to liability of the rotor changing shape or windings shifting.

THE VIBROSCOPE.

This instrument, manufactured by Vibroscope, Inc., N. Y., and described by Mr. Peter Davey in the February 28, 1929, number of "Iron Age," would seem to offer great possibilities, particularly if used in connection with the method of trial balance weights just described as the Westinghouse method. The chief advantage

of the vibroscope is that it provides a sensitive and accurate method of determining the amplitude of vibration of the bearing pedestal.

The Vibroscope makes use of the stroboscopic principle that, by flashing a light on a rotor in synchronism with its rotation, the rotor will appear to be stationary. Figure 24 is a diagrammatic sketch of the apparatus. PA is a head provided with a center similar to that of a tachometer, which center operates the mechanism for flashing the neon lamp NL in synchronism with the rotation of the rotor. The head is also provided with a means of changing the phase angle between flashes and angular position of the rotor. V is the vibroscope proper. It is provided with an arm A in contact with the bearing which transmits the vibrations to an oscillating mirror. This mirror is illuminated by two sources of superimposed light, one a steady white coming from a narrow slit; the other a red light from a similar narrow slit, flashing in synchronism with the neon lamp. Both of these lights are reflected on to a graduated ground glass scale. The width of the band of white light is, of course, a measure of the amplitude of vibration. Suppose now the "high spot" (point 3) of the shaft is in the position shown in Figure 25, *i.e.*, toward that part of the bearing against which the contact arm rests, and that at this instant the red light in the vibroscope flashes, then the reflection on the ground glass scale will be at the extreme right hand edge of the white band as illustrated in Figure 26. The red light appears as a narrow line since it flashes only when a given point of the rotor is at the same position each revolution. NL is also flashing at this instant and the rotor will appear stationary in the position shown in Figure 25. If the phase adjuster is shifted back 90 degrees, the "high spot" (point 3) will have turned through 90 degrees when the neon lamp and the red light in the vibroscope flashes, and will then be at the point occupied by point 5 in Figure 25.

The bearing pedestal will be in its central position, *i.e.*, in the middle of its oscillation at zero amplitude. The flashes of red light must then appear in the middle of the band of the white light as shown in Figure 27. The position of the red light, therefore, indicates the position of the "high spot." Since the "high spot" always *lags* the "heavy spot" as shown in Figure 28, and the angle of lag is 90 degrees at the critical or resonant speed, but

changes very rapidly in this vicinity, the 90 degree angle of lag can be determined quite accurately by slowing the rotor down until the red light changes position rapidly across the scale.

The angle of lag is, of course, required in order that the plane of unbalance be located. This determination would appear to be rather uncertain, but as the amount of unbalance is arrived at by a trial and error method of adding trial balance weights, the angle could also be determined in a like manner. It is understood that use is also made of a force diagram similar to that described above as the Westinghouse method, in which case, of course, the amount of unbalance as well as its angular position, are found more or less directly.

In mass production work, the vibration corresponding to a certain amount of unbalance could be found, after which the unbalance could be found from a calibration curve. This unbalance would be R and R′ of Equations (7), (8), (9), and (10). The angle of unbalance would then have to be determined by means of the red light as described above.

THE GISHOLT BALANCING MACHINE.

A complete description of the Gisholt machine and an explanation of its use may be found in a pamphlet entitled "Dynamic Balancing Simplified," published by the Gisholt Machine Company of Madison, Wis. The general features and principles upon which it works will be covered briefly. Figure 29 is a diagrammatic sketch showing the general arrangement of this machine.

The bed plate B is supported on knife edges K from pedestal P, one end being attached to the flat spring S through the link L, which holds the bed plate in a flexible horizontal position. The rotor turns in bearings M and N, mounted on the bed plate in such a manner that they can be adjusted as to height and longitudinal position. In the headstock H is mounted a shaft carrying a clutch C at one end and balancing disk K at the other, the latter adjustable as to angular position with respect to the shaft at T. The balancing disc carries a balance weight W in a slot so that its distance from the central position can be read on a vernier scale. With the balance weight in zero position, the balancing disc and the clutch assembly are in perfect balance. These parts are motor

driven from below. The headstock is adjustable longitudinally, so that the axial distance between the balance weight and the knife edge is equal to that between the corrction plane A and the knife edge. The amplitude of vibration of the bedplate can be read as the pointer E moves over the graduated scale D.

There are two unique features of this machine which greatly simplifies the calculation of the correction weights; one is that the B correction flange is located directly over the knife edge about which the rotor oscillates, producing the same effect as though the "fixed end" B of previous discussions had moved axially into the correction plane B, *i.e.*, the two planes in effect coincide; the other is the adjustable bearings and headstock by means of which the balance weight W is the same distance from the pivot or the fixed end as is the correction plane A, the effect of which is to make the reference plane or plane of the balance clamp coincide with the correction flange, plane A. These features eliminate forces A_B and B_A, and simplifies the expression for forces A_A and B_B as will be seen below.

$$A_B = \frac{R.a.e.}{L.l} \quad . \quad . \quad . \quad . \quad (7)$$

Since e is zero, the correction A_B disappears.

$$A = \frac{R.a.}{L}\left(1 + \frac{e}{l}\right) \quad . \quad . \quad (8)$$

which reduces to:

$$A_A = \frac{R.a}{l} = k.\ R \text{ (for a given rotor)} \quad . \quad (16)$$

Similarly $B_A = O$, and $B_B = k'.\ R'$ . . . . (17)

The method of procedure follows: a *perfectly balanced* rotor of the type to be balanced is mounted as shown; the balance weight W set out an arbitrary amount, say 25 inch-ounces; the rotor is brought up to a speed just above the resonance frequency of the oscillating parts; the clutch disconnected; the rotor slows down through the resonance frequency; the maximum amplitude of vibration is read on the scale D, say 10 div. From this data the constant k of (16) or (17) above is determined;

$$k = \frac{\text{Bal. wt.} \times \text{Dist. from the Axis (25)}}{\text{Max. Ampl. of Vib. (10)}} = 2.5$$

This relation is based on the principle stated before that the amplitude of vibration is directly proportional to the amount of unbalance for a constant speed.

A rotor of unknown balance is now mounted in the machine; the balance weight W set at zero distance from the axis; the rotor brought up to speed as before; the maximum amplitude of vibration again read, say 12 div. The unbalance in flange A is then:

= (Calibration Constant) × Maximum Amplitude of Vibration
= 2.5 × 12 = 30 inch-ounces.

The method of determining the angular plane of unbalance is unique also. Figure 30, represents the end of a rotor in the plane of the correction flange A. Let the rotor be in perfect balance except for the weight A, which we will assume is 1 ounce at a distance of 1 inch from the axis; the unbalance is 1 inch ounce in correction plane A. Suppose the maximum amplitude of vibration of the machine is 1 div. Now add B, a similar weight of 1 ounce out 1 inch from the axis in the same axial plane; the maximum amplitude of vibration will be 2 div. By placing B 60 degrees to the right of A, the vibration is 1.74 div.; at 90 degrees, 1.41 div.; at 180 degrees, 0 div. From this data the curve, Figure 31, is plotted. Returning to the rotor under balance—the unbalance was 30 inch ounces. The balance weight is set to read 30 inch ounces at an angle of 0 degrees by means of T, Figure 29, and the run repeated. The maximum amplitude of vibration in this condition is noted. Suppose this is now 18 div. The reading with the balance clamp at zero distance from the axis was 12 div. $\frac{18}{12} = 1.5$.

Referring to Figure 31, we find corresponding to 1.5, an angle of 97 degrees. The correct setting for the balance weight is, therefore, 97 degrees to the right or left of its zero position. A trial run will tell which direction is right, for, if set in the correct direction, the rotor should run without vibration.

Due to the fact that so much depends on the relation between the amount of unbalance and the maximum amplitude of vibration of the oscillating system, it seems improbable that results would be extremely accurate. Any remaining unbalance would, of course, manifest itself on the check run, but as there is no way

of knowing whether the weight or the angle of correction is in error, it might take some little time to effect a final balance. The results are, no doubt, sufficiently accurate for all practical purposes.

From the above it is plain that this type of machine is best suited to work of a repititive nature. It is understood that all of the machines built to date are for relatively light work, *i.e.*, for rotors weighing up to 1000 pounds.

THE OLSEN-LUNDGREN BALANCING MACHINE.

Figure 32 is a diagrammatic sketch of the Olsen-Lundgren balancing machine, an illustrated description of which appeared in the "American Machinist" of May 31, 1928, and in "Machinery" for February, 1928. This machine operates on the principle that, at a given velocity of rotation, the centrifugal force at the bearings of the rotor due to unbalance is directly proportional to the amount of unbalance. The rotor is supported in such a manner that this force can be measured directly by means of a calibrated spring.

Referring to Figure 32, the rotor R is carried in bearings at the upper end of the rotor support RS, which in turn is supported from the foundation F by the thin plate KE, the latter acting as a virtual knife edge. The magnitude of the centrifugal force is measured by the micrometer screw MS which is arranged to increase or decrease the tension on the calibrated spring S. The spring S ordinarily holds the oscillating parts against the left hand limit stop L. To find the amount of unbalance, the micrometer screw is adjusted to put sufficient tension on the calibrated spring that the centrifugal force is just sufficient to overcome this tension and force the oscillating parts away from the left hand limit stop. This movement is limited to a matter of 0".001, and is indicated by means of a spark jumping from the pointer P to the dial D, which takes place the instant the contact points C are closed by movement of the oscillating system to the right. The centrifugal force due to the "heavy spot" will be most effective in forcing the oscillating parts away from the left hand limit stop when this "heavy spot" is 90 degrees to the right of the vertical centerline. The circumferential position of the above mentioned spark about the dial D, therefore, indicates the angular position of the "heavy

spot" on the rotor, since the pointer P is synchronized with the revolutions of the rotor, and its position relative to a fixed point on the rotor is known. Theoretically it should be possible to find the amount of unbalance and its angular position for each end at the same time, but it is understood that better results are obtained if one end is balanced at a time.

If the spring S has been calibrated in inch-ounces as is the case, the reading of the micrometer screw MS can be substituted directly in Equations (7), (8), (9), and (10) for R and R′, but it will be noted that the reference plane here coincides with the planes of the bearings, hence, a = L, consequently, these equations reduce to:

$$A_B = \frac{R.e}{l} \quad . \quad . \quad . \quad . \quad . \quad (18)$$

$$A_A = R\left(1 + \frac{e}{l}\right) \quad . \quad . \quad . \quad . \quad (19)$$

$$B_A = R'.\frac{f}{l} \quad . \quad . \quad . \quad . \quad . \quad (20)$$

$$B_B = R'\left(1 + \frac{f}{l}\right) \quad . \quad . \quad . \quad . \quad (21)$$

Some advantages of this machine are that the rotor need not be turned end for end to obtain the unbalance at each end, and that no calibration constant need be worked out for work of different form. On the other hand the calculation of the correction weights and the graphical solution for their angular positions is more complicated than for the Gisholt machine. To simplify this calculation there is furnished with the machine a so-called "transfer instrument," which after a simple setting gives the amount of the correction weight and its angular position for each end for the special case only in which the distance e is equal to the distance f, *i.e.*, the distances between bearings and correction flanges are the same at each end.

SCHENK 60-TON BALANCING MACHINE.

A rather complete description of this machine, manufactured by the Messrs. Carl Schenk, Darmstadt, Germany, will be found in "Engineering" (London), of January 11, 1929. Its most outstanding feature is the arrangement whereby a rotor can be balanced at

its normal running speed as well as at a reduced speed. Due to the possibilities of windings shifting at high speeds, this feature is valuable. Electric motors are provided for most of the adjustments. The rotor is supported much after the manner of the Olsen-Lundgren machine, while the amount of unbalance is found by recording the amplitude of vibration of the bearings at the resonant speed, and using a calibration constant as in the Gisholt machine, but, due to the arrangement, the calculation of correction weights must follow the method outlined for the Olsen-Lundgren machine. The angle of unbalance is apparently determined by chalking the shaft and scribing the "high spot" as the bearing vibrates, running the rotor first in one direction, then the other, and taking the mean between these marks. There are of course many features of this machine which have been omitted.

THE 134-TON WESTINGHOUSE BALANCING MACHINE.

Figure 33 is a diagrammatic end-view of the 134-ton balancing machine developed by the Westinghouse Electric and Mfg. Company and capable of balancing rotors up to 30 feet between bearings and up to fifteen feet in diameter. A complete description of this machine may be found in "Engineering" (London) for September 13, 1929. The rotor shaft R is carried on Kingsbury bearing pads P which are adjustable radially to accommodate shafts of different diameter. The oscillating frame V carrying the bearings is hung from the stationary framework by the thin plate F which acts as a virtual knife edge. The tie rod T prevents lateral motion of V at the bottom, but allows the top free to oscillate. The helical springs S control the amplitude and period of vibration of the oscillating system. Lock screws L act as limit stops when the bearing is allowed to vibrate and provide a means of locking the bearing in the central position when the other bearing is vibrating. Excessive vibrations are damped by the friction damper D. Various details have been necessarily omitted in the sketch.

The Westinghouse method of determining unbalance by means of trial weights is used on this machine, the maximum vibration at the resonant speed being recorded by a dial indicator for various positions of the trial balance weight. After correction weights have been placed in position, generally in a "balancing ring" (cor-

rection flange), the rotor is transferred to its own casing, run under its own power to 20 per cent overspeed, and returned to the balancing machine to correct for any distortions or displacements due to the overspeed. The residual unbalance allowed at the "balancing ring" after test is 1 inch-ounce per 1000 pounds of rotor weight. Unless the balancing rings on all rotors were in the same relative axial positions, such a criterion seems somewhat irrational.

THE AVERY BALANCING MACHINE.

No attempt will be made to describe this unique balancing machine, manufactured by the Messrs. W. and T. Avery, Limited, Engineers, Birmingham, England, and adapted primarily to the balancing of motor car engine crankshafts. Those interested in the details are referred to "Engineering" (London) of July 1, 1927. Its outstanding feature is the design of the balance weight, by virtue of which it may be adjusted as to distance from the axis as well as to angular position while the rotor is running.

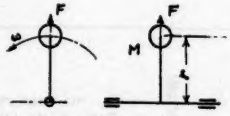


FIG. 1

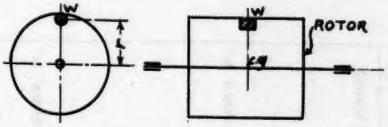


FIG. 5.

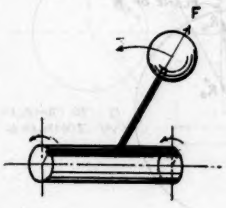


FIG. 2

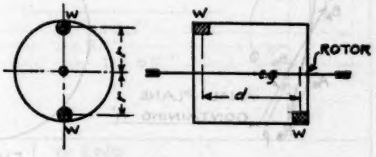


FIG. 6

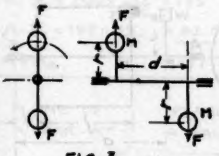


FIG. 3

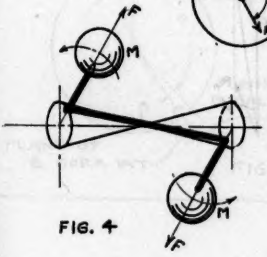


FIG. 4

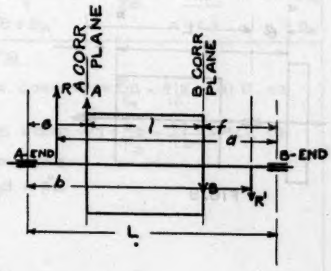


FIG 7

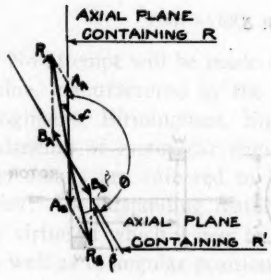


FIG. 8

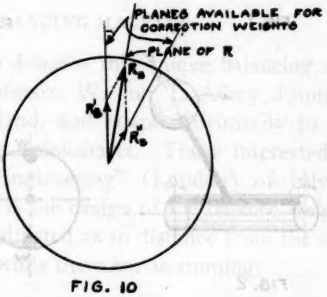


FIG. 10

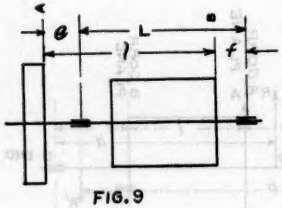


FIG. 9

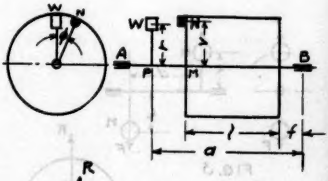
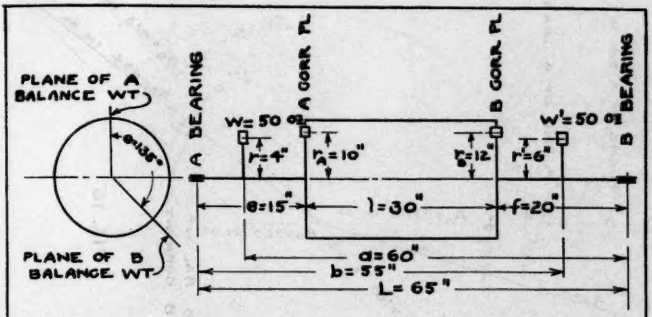
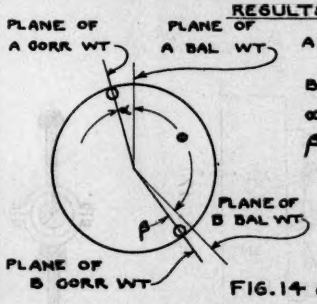


FIG. 11



A-END		B-END	
$W \cdot r = 50 \times 4$	$= 200$ IN.OZ $= R$	$W' \cdot r' = 50 \times 6$	$= 300$ IN.OZ $= R'$
$R \cdot \frac{a}{l} = 200 \times \frac{60}{65}$	$= 185$ " " $= A$	$R' \cdot \frac{b}{l} = 300 \times \frac{55}{65}$	$= 254$ " " $= B$
$A \cdot \frac{e}{l} = 185 \times \frac{15}{30}$	$= 92.5$ " " $= A_B$	$B \cdot \frac{f}{l} = 254 \times \frac{20}{30}$	$= 169$ " " $= B_A$
$A + A_B$	$= 277.5$ " " $= A_A$	$B + B_A$	$= 423$ " " $= B_B$

REGULTS



A CORR WT $= \frac{R_A}{r_A} = \frac{415}{10} = 41.5$ OZ

B CORR WT $= \frac{R_B}{r_B} = \frac{494}{12} = 41.2$ OZ

$\alpha = 16.8°$

$\beta = 72°$

FIG. 14 (a)

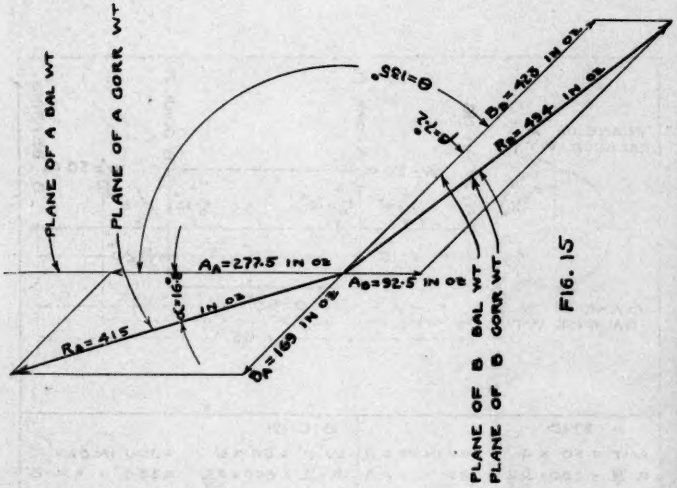


FIG. 15

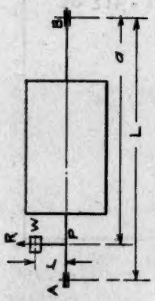


FIG. 14 (b)

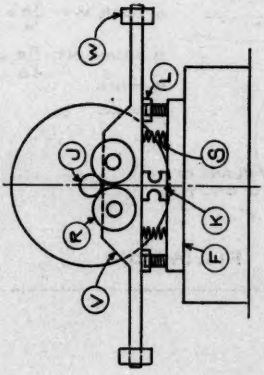


FIG. 17

FIG. 18

FIG. 18

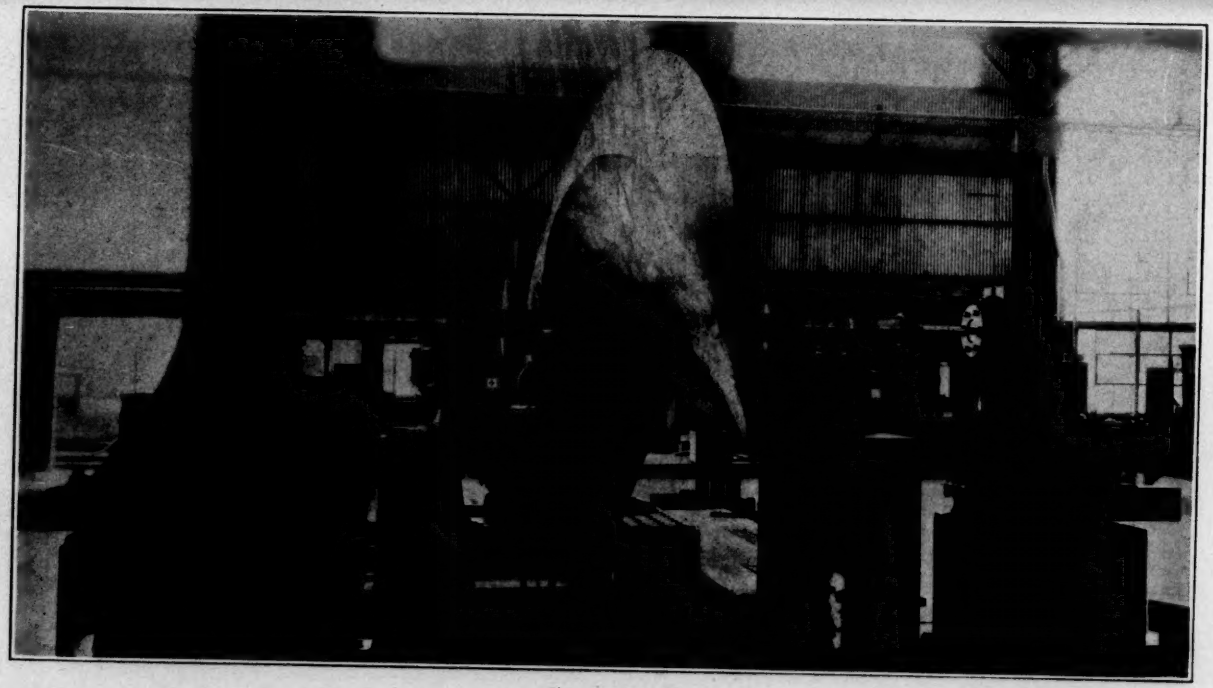


FIGURE 19.

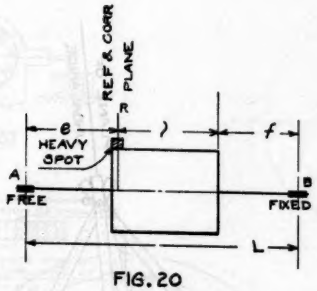


FIG. 20

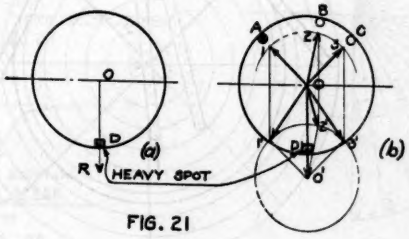


FIG. 21

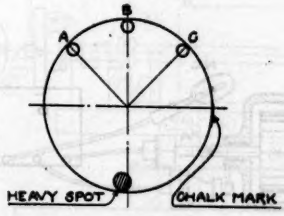


FIG. 22

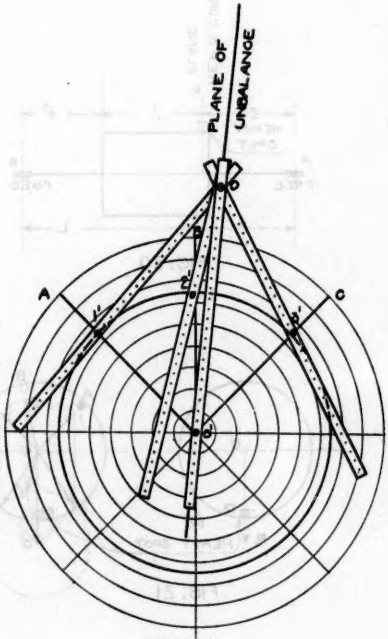


FIG. 23

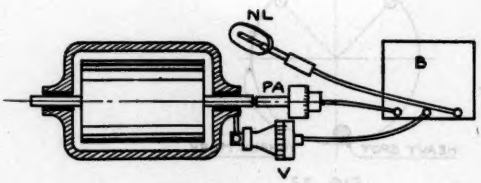


FIG. 24

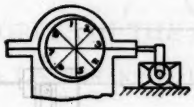


FIG. 25

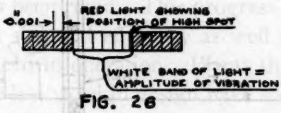


FIG. 26

FIG. 27

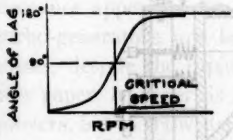


FIG. 28

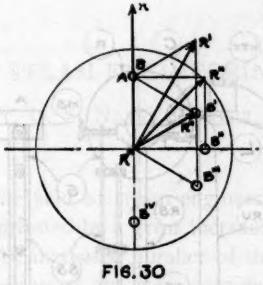


FIG. 30

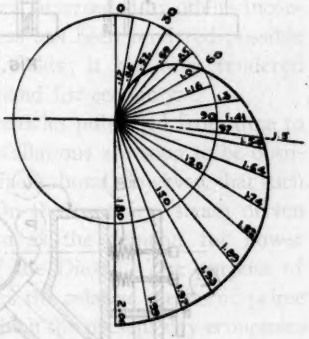


FIG. 31

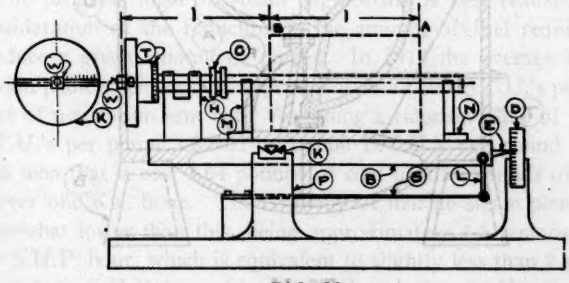


FIG. 29

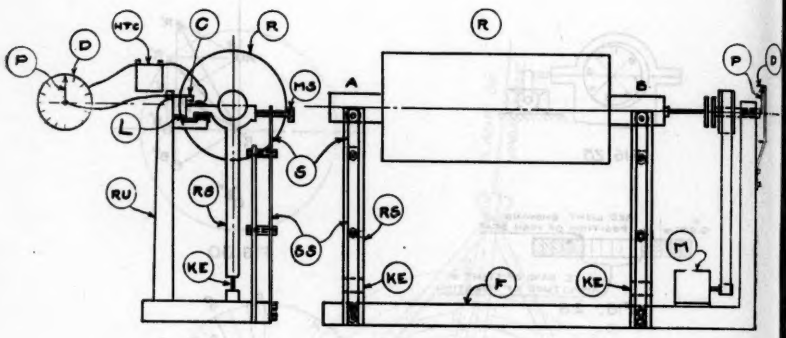


FIG. 32

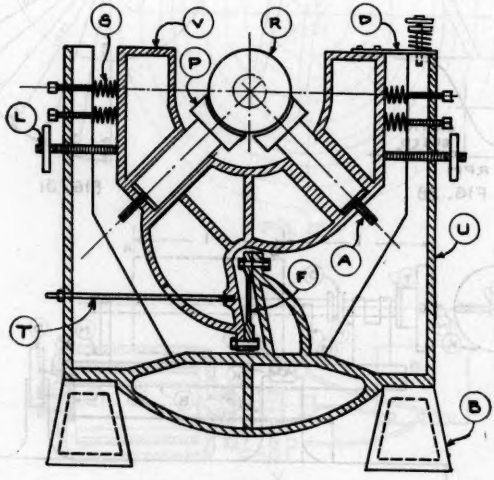


FIG. 33

PROGRESS IN THE FIELD OF STEAM ENGINEERING.

By LIEUTENANT H. G. EBERHART, U. S. NAVY, MEMBER.

For the past decade, progress in the field of steam engineering has been rapid. This progress is manifested by a great increase in the size and efficiency as well as in the increased number of units placed in operation. Plants that a few years ago were considered the last word in design have soon been superseded by others incorporating new features. This progress has been rendered possible by research in hitherto unexplored fields; it has been rendered necessary by the ever increasing demand for economy.

If we are to believe many of the articles published from time to time, it would appear that steam installations are soon to be obsolete. A careful investigation of the facts shows, however, that such does not appear to be the case. On the contrary, steam driven turbo-generators are being installed as the demand for power arises, despite the rapid advance of the Diesel. The purpose of this paper, however, is not to discuss the relative merits of prime movers, but to show the means by which the present day economies in steam plants are effected.

The progress made in steam engineering is best realized by a consideration of the reduction in the amount of fuel required to produce a given amount of power. In 1913 the average central station plant efficiency was approximately 22,500 B.T.U.'s per Kw. hour of net station sent out. Assuming a calorific value of 14,000 B.T.U.'s per pound of coal or 18,000 B.T.U.'s per pound of oil, it is seen that it cost 1.64 pounds of coal or 1.25 pounds of oil to deliver one Kw. hour. The economy of marine steam plants was somewhat lower than this, being approximately 1.5 pounds of oil per S.H.P. hour, which is equivalent to slightly less than 2 pounds of coal per S.H.P. hour. In 1924 the heat units per Kw. hour for the most efficient station in France is given as 22,400 B.T.U., in England as 20,150 B.T.U., and in the United States as 18,030 B.T.U. In 1925 the remarkable performance of the Philo station

of the Ohio Power Company was as low as 13,715 B.T.U. (.98 pound coal, .76 pound oil) per Kw. hour. The equivalent value of these figures is .73 pound coal or .56 pound oil per S.H.P. hour. An increase in plant economy of 39.2 per cent in twelve years is of great moment. Since these results are so greatly at variance with those obtained on board ship, it should be of interest to the marine engineer to see how modern shore practice has achieved them.

In general this gain has been accomplished by:

1. Improvement of the steam cycle.
2. Reduction of losses.

The improvement of the steam cycle is a subject more or less involved with the reduction of losses. It is, however, considered best at this point to show its direct effect on economy; that due to the reduction of losses will be shown later.

Thermodynamics teaches that, in general, the steam plant operates on the Rankine Cycle, which covers the path of the steam from the boiler to the prime mover, to the condenser, and as water, back to the boiler. It is best shown on the P-V diagram (Figure 1).

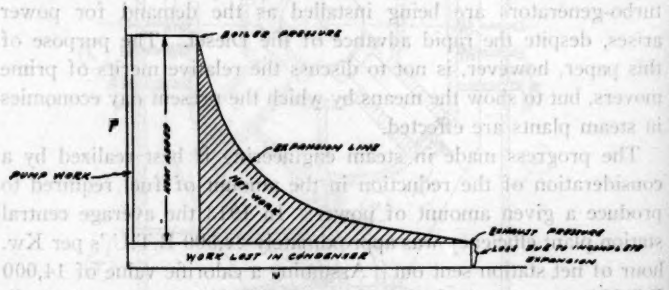


FIGURE 1.

In the theoretical Rankine cycle, the steam expands adiabatically in the prime mover whether it be a reciprocating engine or a turbine. The net work done is the area under the expansion line. It is apparent that with the low admission and high exhaust pressures existing in the early days of steam power, there were two ways of increasing this area and hence, of obtaining more work per pound of steam. One was by an increase in the initial pressure; the other by lowering the exhaust pressure.

It is necessary at this point to look into the past to properly appreciate recent advancement. In 1823 H.M.S. *Lightning* was built at Deptford, England. This was the first steam vessel in the British Navy to take part in naval warfare, and the first for the command of which a commission was granted. She may be taken as an example of up-to-date design of that period. It is doubtful if any shore stations were in existence which were better. Her engines of 100 nominal horsepower, had two cylinders 40½ inches diameter and 4-feet stroke. Her boilers supplied steam at but 2 to 3 pounds pressure. Salt water was used as feed.

The first step taken to increase steam economy was in the direction of higher steam pressure. Records show that almost forty years later Napier constructed engines for the Cunard Company's *Scotia* which had two cylinders of 100 inches diameter and 12-feet stroke. They were supplied by steam having the enormous initial pressure of 20 pounds per square inch.

As steam pressures rose, feed water troubles began to appear. Condensers resulted. From this time on, economy improved steadily by an increase in the initial and a lowering of exhaust pressures until the early part of the present century found the condensing, reciprocating engine, 150 pounds pressure Scotch boiler combination, definitely established.

During this period of transition great impetus was given to the art by the increase in the general use of electricity. This gave rise to the central station which in turn sought ways and means of producing power more cheaply. Since the lowering of the exhaust pressure gave such great rewards, much of the effort was expended in this direction. It was not long, however, before the reciprocating engine reached the limit to the vacuum it could efficiently use. Because of the high specific volume of steam at very low pressures, it was found to be impracticable to utilize the benefits of lowering the exhaust pressure below that corresponding to a vacuum of 26 inches of mercury. This greatly accelerated the development of the steam turbine since it can utilize to great advantage the benefits accruing from the low exhaust pressure that modern condensing equipment can and does produce. Because of this, the reciprocating steam engine alone, is unable to compete in the production of power on a large scale. To illustrate this point,

it may be of interest to know that in 1925 the Interboro Rapid Transit Company, of New York City, had, in addition to several 30,000 Kw. turbo-generator units, several old reciprocating engine units. These were relics of an earlier period. Despite the fact that their output had been increased nearly 50 per cent by altering them to exhaust to low pressure turbines instead of to a condenser, their steam consumption was so high that they were run only on peak loads of comparatively short duration. The station base load was carried day in and day out by the 30,000 Kw. turbo-generator units. The past twenty-five years has seen massive reciprocating steam engines producing perhaps 10,000 Kw. replaced by turbines which by virtue of their greater economy have grown in numbers and size until the year 1929 witnessed the installation at Hell Gate of a turbo-generator unit with a rated load of 160,000 Kw.

While exhaust pressures were being lowered, progress was made in the raising of initial steam pressures. The Scotch boiler, due to its construction, reached its upper limit at about 200 pounds per square inch. Then as the need for greater pressure became manifest, the water tube boiler came into general use. All modern, high pressure, high capacity boilers are of this type.

Condensers soon reached the low limit of pressure that it is practicable to maintain. Given a sufficient supply of circulating water of the requisite temperature, it is possible to build a condenser that will maintain practically any specified vacuum. The turbine, however, beyond a certain designed point, usually 1 pound absolute at rated load, suffers the limitations in its low pressure stages of the reciprocating engine relative to the high specific volume of the steam. Since under given conditions it costs approximately twice as much to build a condenser capable of maintaining, at the rated load of the turbine, a vacuum of 29 inches Hg. as it does to build one capable of maintaining a vacuum of 28 inches, this latter figure is usually specified. The turbine is designed accordingly. Since, as before stated, condensers have reached their practicable low pressure limit, today finds the entire world of steam engineering pushing on up into the hitherto unexplored regions of high pressure steam in an attempt to improve the efficiency of the cycle.

According to the first law of Thermodynamics, heat may be converted into work. If this were not true there would be neither steam nor internal combustion engines. We must, therefore, take as a premise that steam does work by virue of the heat it contains. Steam, at any given pressure, has a certain Heat Content expressed in B.T.U. per pound. Heat Content at any given pressure, depends on quality and superheat, but to establish a point of reference, for the purpose of discussion, saturated steam will be used in explaining the reason for the use of high initial pressure.

Saturated steam at any specified pressure has a definite temperature and Heat Content. It is important at this point to consider this relationship quantitatively as shown by the following table.

TABLE OF HEAT CONTENT OF SATURATED STEAM.

Pressure Lbs. Per Sq. In. Absolute	*Heat Content B.T.U. Per Pound*
10	1143
60	1177
150	1193
200	1198
300	1204
400	1208
500	1210
1000	1188
1200	1177

From the steam tables it is seen that the total heat of steam increases very slowly with an increase in pressure up to about 500 pounds per square inch, remains stationary till at about 600 pounds it decreases slowly. At a pressure of 1200 pounds it is the same as at 60 pounds per square inch. It is also to be noticed that within the range of the pressures shown, the total variation in Heat Content is very small. Its maximum value is 1210 B.T.U.'s, the minimum at 10 pounds pressure is 1143; a change of 67 B.T.U.'s or approximately one-half of one per cent. If this were the whole story, there would be so little gain in economy by an increase in pressure that even moderately high pressures would scarcely compensate for the added cost of the boilers involved in

its generation. It is known, however, that a greatly increased amount of work can be obtained from a pound of steam if its pressure is raised decidedly. This is due, not to the change in its total heat, but to a property of steam known as Entropy. This may be defined as being "a measure of the availability of the energy of the steam for doing work." As the pressure is raised, the Entropy increases, hence its available energy increases. This means simply, that the higher the pressure, the greater the amount of available energy there is in a definite amount of steam. Therein lies the value of high steam pressure. This is best shown by the following diagram; viz.,

SATURATED STEAM. VALUES OF AVAILABLE ENERGY *vs.* PRESSURE.

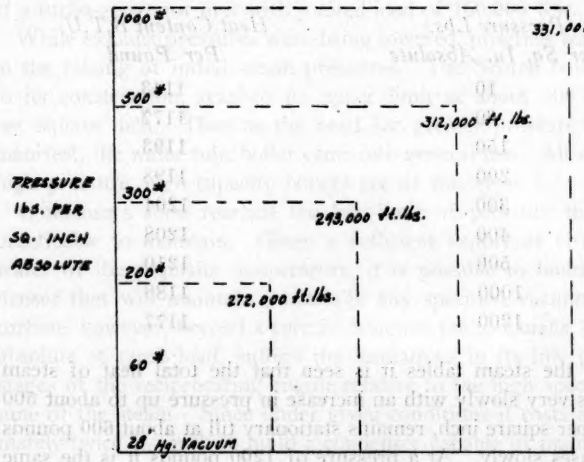


FIGURE 2.

This increase of the available energy of the steam is somewhat analogous to the raising of a weight from one level to another that is higher and thereby increasing its potential energy. The difference in the two cases being that in the raising of the weight a definite amount of work must be done, whereas in the raising of the steam pressure but little or no increase in fuel is necessary. All that is required is to add the slight amount of heat indicated by

the difference in Heat Content between that of the steam at one level of pressure to that of a higher one. It should be noted at this point that the Heat Content of steam at 60 pounds per square inch is the same as that at 1200 pounds, hence no additional heat would be required to accomplish the change between these levels. With these facts in mind, it is of interest to investigate the theoretical saving in raising the pressure from 200 to 1000 pounds per square inch.

The increase in available energy per pound of steam is,

$$331{,}000 - 272{,}000 = 59{,}000 \text{ foot pounds}$$

This represents a gain of 21.69 per cent. In addition, the saving in fuel, based on relative Heat Contents is

$$1{,}198.5 - 1{,}187.6 = 10.9 \text{ B.T.U.'s, or .8 per cent.}$$

The total saving, being the sum of these, is 21.69 per cent plus .8 per cent or 22.49 per cent. Thus for a given output of power, the fuel required would, theoretically, be reduced by this amount.

In practice, however, certain inherent difficulties arise which militate against the complete utilization of this theoretical gain. These obstacles may be considered to be physical and commercial. Quite naturally they are related and are such that commercial considerations govern the limit of the present and subsequent steam pressures in general use. Of the two it may be said that the commercial considerations are the most subject to change. It is believed that when the economic considerations warrant it, the physical difficulties will be solved.

In order to understand the physical obstacles in the way of the immediate adoption of pressures in excess of 400 pounds, it is necessary to consider the following points, *viz.*,

(1) Saturated steam at any given pressure, has a definite temperature. As the pressure is increased the temperature rises. At 100 pounds per square inch absolute, the temperature corresponding is 327.8 F.; at 660 pounds, it is 486.6 F.

(2) As the pressure increases the latent heat of evaporation decreases. Steam expanding adiabatically in the prime mover gives up a portion of its heat in the performance of work. Because of this, it becomes moist to a degree depending on the amount of

work done and its latent heat of vaporization, consequently, as the pressure is raised, the moisture content of the steam in the prime mover increases.

The effects of these characteristics of steam give rise to most of the major obstacles in the way of the general adoption of high steam pressures.

The thermal efficiency of a heat engine depends largely upon the difference in temperature of the working medium at admission and exhaust. The temperature at exhaust involves no difficulty since it is low and is definitely established by the arbitrarily chosen exhaust pressure. Any increase in the temperature drop through the engine must, therefore, be accomplished by an increase in the initial temperature. This may be done by higher pressure, by superheat, or by both. Unfortunately, the upper limit of temperature is established by the physical properties of materials employed.

At high temperatures, materials behave in a manner resembling viscous fluids, that is, deformations continue for long periods of time. There is a certain stress at which creepage will start after a certain time for each material and temperature. This stress is defined as the creep limit. Ordinary high grade materials used in steam turbines have their physical strength gradually impaired at temperatures above 570 degrees F.

In addition to the ultimate effect of high temperature on the material employed, there is an immediate weakening or lowering of its tensile strength if a certain point is reached. Of course turbine blading is made from various alloys which are comparatively heat resisting. At the present stage of the art, however, boilers and piping are made of mild steel. At temperatures above 750 degrees F. the tensile strength of mild steel falls off sharply. In boilers, particularly, it is not feasible to increase tube thickness to compensate for this decrease in strength without due regard to the problems of heat transfer involved. As a result of the limitations of the metals used in the boiler and turbine an initial steam temperature of 750 degrees F. is regarded today as the safe upper limit.

An initial steam temperature of 750 degrees F. may be obtained by the use of saturated steam generated at the corresponding pressure. Since steam at this pressure becomes unduly wet at exhaust

excessive erosion of fixed and moving blades results. In addition, a marked decrease in turbine efficiency results from its high moisture content. With saturated steam of moderate pressure at the throttle, friction loss across blades and in nozzle is approximately 20 per cent of the total available energy of the steam. As the pressure is increased this loss rises decidedly unless steps are taken to prevent excessive moisture. One alternative is superheat.

If at any given pressure, heat is applied to saturated steam not in the presence of water, the first effect is to insure its absolute freedom from entrained moisture and next to elevate its temperature. Upon the continued application of heat, it will rise in temperature approximately 2 degrees F. for each B.T.U. added per pound of steam. It is then Superheated Steam. Upon adiabatic expansion it will give up this superheat to perform work before it gives up any of its latent heat of vaporization. This tends to prevent excessive moisture in the working medium. The use of superheat is therefore resorted to, to increase turbine efficiency. It must be borne in mind, however, that 750 degrees F. represents the present upper limit of temperature, consequently a decision must be made in all cases as to whether a pressure of 400 pounds with high superheat or 1000 pounds with little or no superheat is to be employed. Theoretically the gain resulting from superheat is that resulting from a greater heat content per pound of steam upon admission. Actually its effect on turbine efficiency is much greater because of the decrease in blade friction due to its use. In the case of high pressure steam the gain resulting from its use is greater than in the case of steam of moderate initial pressure. The following table shows the minimum results to be expected. The effect of superheating steam at a pressure of 250 pounds is as follows:

Degrees F of Superheat	Foot pounds energy per pound of steam	Net theoretical gain per cent	Actual gain per cent.
0	298,000	0	0
50	308,500	.64	1.11
300	347,000	2.90	10.45

Commercial stations, realizing the benefits to be obtained, endeavor to obtain the maximum amount of superheat possible without exceeding a final temperature of 700 degrees F. Manifestly with a steam pressure of 1000 pounds the amount of superheat cannot be great so with high initial pressures, modern practice has adopted reheat.

This is the process of interrupting the steam in its expansion through the turbine at a point of predetermined moisture content. The steam at a somewhat lowered pressure is led through the reheater where it absorbs additional heat and becomes superheated. It is then returned to the next lower stage of the turbine where its expansion continues. By reducing blade friction to a minimum its benefits are great. It has, however, the disadvantage of increasing the weight, space, and of greater commercial moment, cost.

In addition to the basic physical obstacles to the general adoption of steam pressures above 600 pounds, certain mechanical difficulties exist. These are the difficulties of manufacture, maintenance and operation of boilers, pumps and valves to accommodate these pressures. These, in several cases have been successfully met. The cost involved, however, exerts a profound influence on projected design.

In any consideration of this problem it must be remembered that the real function of power station engineers is to deliver power on the station busbars at the lowest possible cost per kilowatt-hour. This cost is composed of four major elements: (*a*) Operating labor and superintendence; (*b*) maintenance; (*c*) fuel cost; (*d*) fixed charges on the investment. Of these, the fuel cost and fixed charges are the largest items. They are at present, approximately equal, hence any increase in fixed charges must result in a corresponding saving in fuel costs. It is needless to point out that if fuel should advance materially in price, additional fixed charges to accomplish increased economy would be warranted.

Subject to the foregoing limitations, the net result at present, is a decided gain in commercial economy due to an increase in pressure up to approximately 400 pounds. From that point to about 600 pounds, the gain, commercially, is slight. Beyond that point, present commercial efficiencies are reduced.

While commercial practice has been steadily pushing pressures upward and is pointing still higher, it is believed that 600 pounds can be considered as somewhat above the present average. An exception to this is the Weymouth Power Station of the Edison Electric Illuminating Co., of Boston, which has installed in addition to a basic plant of 60,000 Kw. capacity, operating on 350 pound pressure, a 1200 pound steam plant having a capacity of 15,000 Kw. A relatively new 78,000 Kw. turbo generator with an initial pressure of 600 pounds is operating successfully in Chicago. Many other examples exist.

Despite the thought and effort being applied to the improvement of the steam cycle by the increase of initial pressure, a point has been reached where further gain will be difficult of attainment if the straight steam cycle is adhered to. It is estimated that subject to present limitations an economy of 13,000 B.T.U. per Kw. hour is about the best that can be obtained for a station designed to operate on the straight steam cycle with a single stage of reheating. A possibility of obtaining increased economy is that presented by the use of the combined mercury vapor-steam cycle. By the employment of this cycle on an experimental scale in the Dutch Point Station of the Hartford Electric Light Co., an indicated economy of 12,000 B.T.U. per Kw. hour was obtained. Further development along these lines, it is estimated, will give an economy of approximately 10,000 B.T.U. per Kw. hour. The mercury vapor-steam cycle is so great a departure from previous practice and so filled with possibilities that despite the fact that it is not yet in general use, a brief description follows.

This cycle represents an attempt to work in the field of wide temperature differences, without resorting to high pressure and superheat with their attendant difficulties. It depends for its success on the physical characteristics of mercury which boils at a temperature of 800 degrees F. at a pressure of 70 pounds per square inch. The cycle is shown diagrammatically by Figure 3.

It is seen by the foregoing Figure that mercury is evaporated at a very high temperature in a low pressure, "water tube" boiler. The vapor is then expanded through a specially designed turbine and exhausts to a condenser. This condenser is in reality a medium pressure boiler. Its circulating water which is fresh is transformed

into steam at a pressure of 250 pounds per square inch by the heat of the mercury vapor at exhaust. The steam so generated is then led to the steam mains of the plant and is used to run ordinary steam turbo generators.

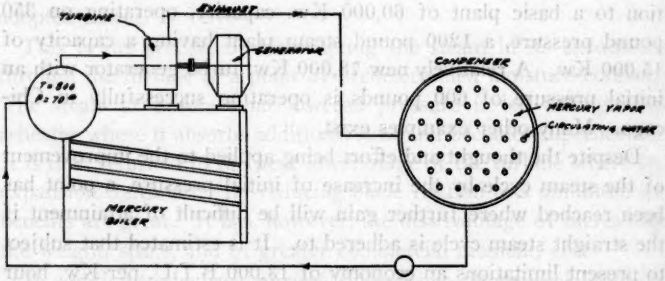


FIGURE 3.

In this cycle, several points are worthy of note. The great temperature difference between the mercury vapor at 800 F., and the steam at exhaust accomplished at relatively low pressures, is of course the most important. The almost complete utilization of heat in the portion of the cycle containing mercury is highly contributory to its efficiency. Because of the low pressure at which it is generated, the mercury vapor has a low spouting velocity, hence the mercury turbine will run at about one-third the speed of a steam turbine. Since mercury carries nothing in solution, there is no problem of corrosion or scale formation in the boiler tubes. It has been found that when mercury is substituted for water that a higher rate of heat transfer results. Higher boiler efficiency obtains.

From a consideration of the various factors involved, it is believed by many that the straight steam cycle will eventually give way to some form of the combination vapor cycle in order that its greater economy may be realized.

As previously stated, present economies are obtained by a reduction of losses as well as by improvement of the cycle. Major losses are encountered in the boilers, turbines, condensers and the use of steam driven auxiliaries. Due to the stress of competition a great deal of effort is expended towards their reduction.

With a given installation it is recognized that it is to the boiler operation that most of the gain or loss in economy can be traced. Commercial plants pay great attention to this particular and keep elaborate records of the CO_2 content of flue gases, stack temperatures, steam pressures and temperatures, feed water temperatures and smoke. Cinders are caught and all ashes are washed to recover unburned carbon. The saving effected by this last item amounts to about 1 per cent of the total fuel used. This in the case of the Interboro Rapid Transit Co. of New York which has a daily fuel consumption of 2000 tons, amounts to a sizeable item. It is not the purpose of this article to deal with boiler operation except to show that by these means faulty operation is checked at once.

Fundamentally boiler design has not changed greatly except that boilers have grown in size. Certain modifications as to furnaces and drums have occurred but they are still water tube boilers. A great deal of work has been done along the line of superheater location in order to obtain a constant degree of superheat at all ratings. Superheaters were first moved from the region of lowest gas temperatures to the region of maximum temperatures exposed to radiant heat. This location was found to be unsatisfactory and so they are now generally located just above the second or third row of tubes. This gives a high degree of superheat which is fairly constant for all ratings.

Research in heat transfer has established the fact that the greatest rate of exchange of heat occurs when the flow of the gases is at right angles to the tubes, and least when the flow of gas is along the tube. In view of this, no boiler can lay claim to good design unless its tubes are staggered to avoid stratification of the gases and has the flow of gas across them. The velocity of the water in the tubes is also a function of heat transfer. Rapid circulation increases boiler efficiency by increasing the amount of heat absorbed. It is also essential at high ratings, to insure that the generating tubes do not become too dry and overheated. This is accomplished by proper baffling and the placing of steam drums in the correct locations.

The air for combusion carries large quantities of heat up the stack even with the best of control. When large amounts of excess air exist, the loss is very high. To reduce it to a minimum, an

attempt is made to have the gases leave at a temperature as near to that of the steam as possible. One expedient is to install economizers which absorb heat from the flue gases and impart it to the feed water after it has left the feed heater. These are heavy, bulky and expensive and do not appear to be particularly adapted to marine use. A new line of attack on this problem is coming to the fore. It is a means of preheating the air for combustion by means of the waste heat of the flue gas. This system has the further advantage that since the temperature of the air entering the furnace is nearly equal to its temperature, on leaving the flues, excess air would no longer figure so largely as a loss.

In addition to the effort expended on the design and operation of boilers in an effort to secure economy, is the work done on turbines. To increase their efficiency is of course to reduce their losses. The solution of the problem lies in this direction. Turbine losses are as follows:

(*a*) *Windage and friction.*

Windage is the loss caused by the churning of the steam by the blades as they revolve at high speed. This is a function of blade size, peripheral speed and the density of the medium. A realization of this demonstrates the necessity of high vaccum as a means of reducing this loss.

Friction is the loss resulting from the passage of steam across the blades and through the nozzles. This has been found to increase with the moisture content of the steam. For this reason superheat or reheat is essential. It has also been found that the design of the blade, *i.e.*, its shape, angle of entrance, angle of exit, and smoothness of surface have a decided bearing on turbine efficiency. The present practice among the best designers of large impulse turbines is to carefully machine and finish all blading.

(*b*) *Packing Loss.*

This is the loss occasioned by the leakage from one stage to another. It amounts to about 1 per cent and has practically reached its lowest limit.

(*c*) *Leaving Loss.*

As steam travels from the high pressure to the low pressure end of the turbine, it must, regardless of its velocity across the blades,

have an axial component of travel sufficient to allow the requisite amount of steam to pass through the turbine. Whatever axial velocity the steam has as it leaves the last row of blading, is lost insofar as its being transformed into useful work is concerned. This loss is known as the "Leaving Loss."

The blading in the lowest pressure stage is limited in length by the speed of rotation, hence the effective area of the spaces between the last stage blading is established. This in turn limits the amount of steam that can be put through any given turbine for a definite leaving loss. Practice establishes the low limit of this axial velocity at about 500 feet per second. This loss amounts to about 4 per cent of the available energy of the steam and is obtained by the formula, K.E. $= \frac{WV^2}{2g}$, where W = weight of steam, lbs. V = axial velocity, feet per second. Since it is proportional to the square of the axial velocity any reduction of this component contributes decidedly to turbine efficiency. Its reduction is a matter of design. A consideration of the following features illustrates the methods employed towards securing its reduction.

The arrangement shown in Figure 4 is typical of many destroyer turbines. It is one in which the leaving losses are high. They

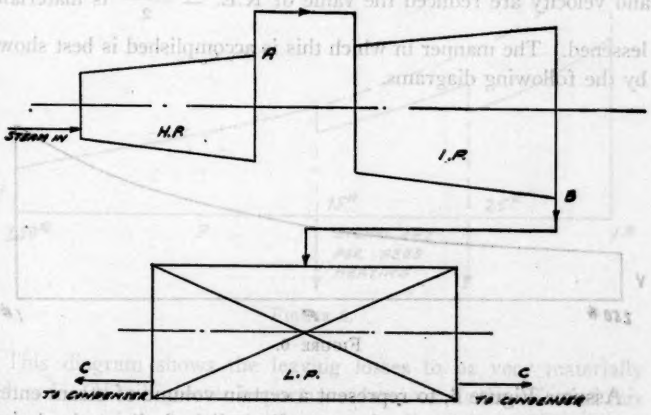


FIGURE 4.

occur at A, B and C. Recent design reduces this by providing an uninterrupted flow of steam through the turbine as shown by Figure 5.

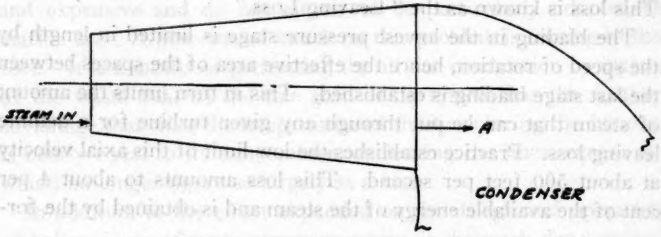


FIGURE 5.

By this arrangement the leaving loss occurs only at A. It may be lowered still further by reducing the amount of steam leaving the last row of blades. This is done by "bleeding" the turbine at one or more points prior to exhaust and using the steam so abstracted for heating feed water or other purposes. Any reduction in the amount of steam leaving the last row of blading will enable the designer to reduce the axial velocity of exit. Since both mass and velocity are reduced the value of K.E. $= \frac{MV^2}{2}$ is materially lessened. The manner in which this is accomplished is best shown by the following diagrams.

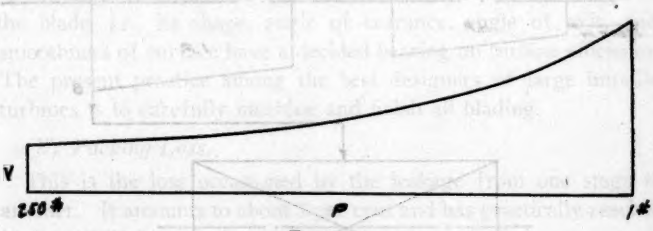


FIGURE 6.

Assume Figure 6, to represent a certain volume of steam entering the turbine at 250 pounds, expanding adiabatically, and exhaust-

ing to the condenser. If, at some one of its lower stages, auxiliary exhaust is introduced, the condition becomes as is shown in Figure 7.

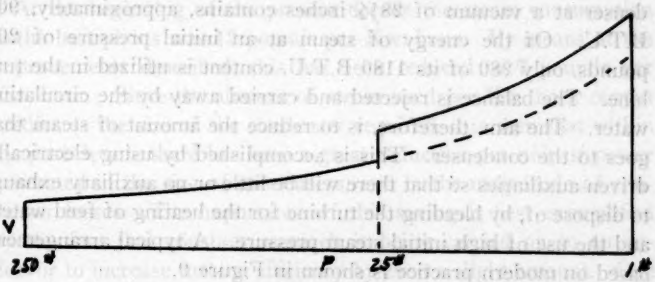


FIGURE 7.

This is an arrangement commonly found in marine practice. It will be seen that this causes a congestion of the last rows of blading when the turbine is operating at rated load, thereby increasing the frictional and leaving losses. While this is common procedure it is poor from a standpoint of economy. Better practice is that shown by Figure 8.

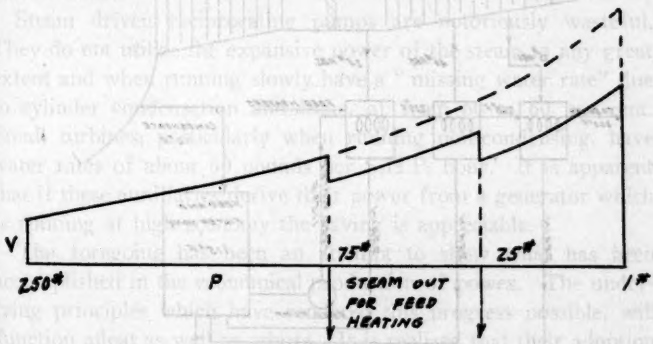


FIGURE 8.

This diagram shows the leaving losses to be very materially reduced. Of still greater importance, however, is the effect of this arrangement in reducing the Loss to the Condenser.

One of the greatest of losses in the steam cycle is that resulting from the large quantity of heat rejected to the condenser in the form of exhaust steam. Each pound of steam entering the condenser at a vacuum of 28½ inches contains, approximately, 900 B.T.U. Of the energy of steam at an initial pressure of 200 pounds, only 280 of its 1180 B.T.U. content is utilized in the turbine. The balance is rejected and carried away by the circulating water. The aim, therefore, is to reduce the amount of steam that goes to the condenser. This is accomplished by using electrically driven auxiliaries so that there will be little or no auxiliary exhaust to dispose of, by bleeding the turbine for the heating of feed water, and the use of high initial steam pressure. A typical arrangement based on modern practice is shown in Figure 9.

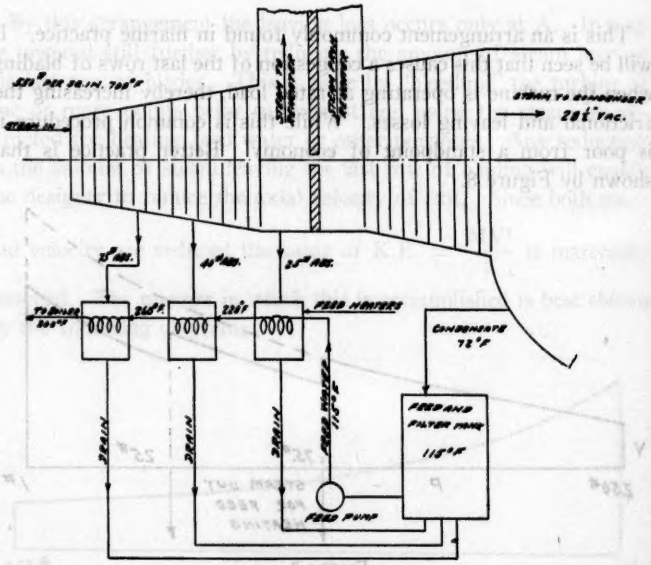


FIGURE 9.

By this arrangement the heat of a certain portion of the steam is completely utilized and the efficiency of the cycle is correspondingly increased. This coupled with a reduction in the amount of

steam entering the prime mover initially by virtue of using high pressure, high temperature steam and improved turbine design, reduces materially the amount of exhaust to the condenser.

At the beginning of the present decade the steam consumption in the best plants was 12 pounds per Kw. hour. By means of the improvements described, this has been reduced to 7 pounds per Kw. hour. Since the amount of steam used per Kw. hour output predicates the size of the condenser required, it is seen that within the past decade there has been a great reduction per unit of power generated in the size of condensers, and in the magnitude of condenser losses. The importance of this saving is best exemplified by the fact that constant research is being carried on in an endeavor to increase turbine efficiency by even a slight amount. One of the largest turbine manufacturers in the United States considers the expenditure of one million dollars in research justified, if it results in an increase of 1 per cent in the efficiency of their product.

In the commercial world, electrically driven auxiliaries are in general and constant use. They are considered rugged and reliable and by virtue of their high economy have largely displaced steam driven units.

Steam driven reciprocating pumps are notoriously wasteful. They do not utilize the expansive power of the steam to any great extent and when running slowly have a "missing water rate" due to cylinder condensation and leaks, of from 50 to 60 per cent. Small turbines, particularly when running non-condensing, have water rates of about 50 pounds per S.H.P. hour. It is apparent that if these auxiliaries derive their power from a generator which is running at high economy the saving is appreciable.

The foregoing has been an attempt to show what has been accomplished in the economical production of power. The underlying principles which have rendered this progress possible, will function afloat as well as ashore. It is realized that their adoption on board ship offers special problems relative to weight and space. Naval vessels on the other hand, are not concerned with commercial efficiency since cruising radius and reliability are of far greater moment than cost.

In conclusion it is desired to point out that a large vessel need not, in general, differ fundamentally from a large shore station and so there is no real reason why with similar design, comparable fuel economies cannot be obtained. Obviously it involves electric drive, this we have; a more generous use of electrically driven auxiliaries, this is practicable; high pressure and high superheat coupled with reheat, this involves no unsolved problems.

That high economies can be obtained on board ship, by the use of modern ideas and equipment, is firmly believed in by the engineers of one of the largest concerns in the United States. This opinion is so strongly held that they guarantee a reduction in the fuel bill of one of our biggest liners of 600 tons a day, *i.e.*, from 1,000 tons to 400 tons each 24 hours.

Modern liners built within the past five years show the effect of the adoption of these features in greatly increased fuel economy. The ship of tomorrow will doubtless embody still greater improvements. It further appears that the steam driven turbine because of its simplicity, small weight and improved economy, will continue for some time to be the prime mover on ships where reliability, high power, maneuvering ability and weight of machinery within reasonable limits is of vital necessity.

STEAM TURBINE LUBRICATION.

By S. M. Riis, Member.

Note: This discussion applies only to direct drive steam turbine circulating system lubrication.

EMULSION.

Some years ago when the steam turbine had established itself in a technical world as a prime steam mover of reliability and when it began to replace the reciprocating type for economic, as well as for comparative simplicity of operating reasons, I became interested in the problems of lubrication in connection with proper and continuous operation of such installations from point of view of the lubricant itself, without consideration as to technical and construction aspects of the machinery. It then was apparent that the greatest drawback was that the various lubricating oils recommended and used gave great deal of trouble by reason that there was the formation of emulsions in the lubricating system. The most important factor to my mind was then and is now to find means which will guard against such emulsion by selecting and supplying oils which will cut to absolute minimum any chances of such emulsions without taking into consideration the technical construction of the turbine or its circulating system which vary to certain extent in the various types and makes of steam turbines.

It has been noted that the mean bearing temperatures of various types of turbines vary, notwithstanding the fact that exactly the same brand of oil was used in its circulating system. Even in case of two twin turbines of exactly the same size, same voltage, same revolutions, installed by same engineers and manufactured by the same company, operated exactly under same conditions and loads, each one showed a different mean bearing temperature even though the lubricating oil was also exactly of same specifications, manufactured in one lot, and the water used was also the same.

The difference of bearing temperatures, after several years of practical experimenting under actual working conditions, was

traced by me to the fact that the oils supplied emulsified more in the circulating system of one turbine than in the other. Yet the tests applied to oil in question as to its capability to separate from water, or demulsification test, was the same in both cases.

Why does the same oil, similarly tested and manufactured in one lot, emulsify more in one case, under identically similar operating conditions, and less in the other case? Reasons for this I have ascertained after 12,000 working hours with a certain brand of oil in two similar turbines.

The opportunity to carry out such experiments under actual working conditions was made possible when I was assigned as Lubricating Manager, Standard-Nobel Co., in Poland (subsidiary of Standard Oil Co. of N. J.) in 1927. At that time there had never been manufactured a successful turbine lubricant in Poland in Polish refineries from Polish crude. Basing my ideas on past studies regarding this question, I instructed our refinery to manufacture turbine oil answering the general specifications as generally adopted for high speed stationary turbine lubrication (visc. Eng. 2.75 at 50 degrees C.), with the exception that in the values of temperatures for emulsion test (settling) I indicated the temperature 100 degrees (officially used in Germany) 80 degrees C. (officially used in Poland) 54.4 degrees C. (officially used in U. S. A.) and at 40 degrees C. (my own formula), requiring that the settling should be identical in all these temperatures after one hour. To my surprise I found that the settling was not identical or uniform, that the oil, settling at higher temperatures, settled quicker than at lower. This might have some connection with the properties of Polish Borislaw crude, which, however, I doubt. Therefore I came to the conclusion that if oil does settle on emulsion test at higher temperatures quicker—then, if the mechanical construction of the bearings and alignment is such that lower bearing temperature is possible, and if the oil is such that it separates readily from water at lower temperatures, the maintenance of low bearing temperature is also possible. This was eventually proven in practice.

For instance, suppose we take oil which readily (in one hour) separates from water at 80 degrees C. or at 55 degrees C. but not at 40 degrees C. and the desired bearing temperature is 45 degrees

C., then the temperature of the oil in the circulating system should be maintained below 45 degrees C. and if the oil emulsifies and does not readily separate from water at temperature below 45 degrees C. there at once is a subsequent loss in lubricating power, and such emulsion will cause the formation of soluble sludge which is further converted by the action of heat to insoluble sludges. This permanent sludge will partially clog the oil pipes and oil ducts as well as cause an increase in the fluid friction of the lubricant. The result—the bearing temperature goes up to the temperature under which the oil begins to readily separate from water. The desired low bearing temperatures cannot be maintained.

It has been proven to my satisfaction that if the emulsification test is carried out at 40 degrees or below, the oil will not emulsify at higher temperatures. In Germany and in Poland at present there is a practice of carrying out emulsion tests at temperature of 100 degrees C. with steam instead of water. This is done under assumption that the wet steam in the turbine is about 100 degrees C. and if under such conditions the oil does not emulsify permanently it will not emulsify under actual working conditions. My opinion is contrary to this assumption inasmuch as the oil while actually settling, goes through the process of demulsification or separation from water or heavy moisture in a temperature much lower than the lowest bearing temperature possible. Therefore the oil must be tested for emulsion at lower temperature than any bearing temperature which might be desired.

The practical tests to prove my contention have been carried out in lubricating medium-large size turbines under actual working conditions for periods of 12,000, 10,000 and 8,000 working hours. The mean bearing temperatures which were maintained without any traces of emulsion on the three turbines has been 49 degrees C. during summer heat and 46 degrees C. during winter.

The turbines were manufactured by:

1. Society General .....16,000 Kw. 3500 R.P.M.
2. Brown Boveri .....16,000 Kw. 3600 R.P.M.
   Warsaw City Electric Light—Poland.
3. Bruno Morawski, Chekoslovakia,
   (a new type turbine) .....10,000 Kw. 3600 R.P.M.
   Firley Cement Works, near Lublin—Poland.

All these three turbines were newly installed with our 275 Turbine Oil and no other oils have ever been tried in them. The water at Lublin was several points harder than the one at Warsaw. The operating results as to lubrication were practically identical in all three cases.

FOAMING.

There is always a certain amount of foaming in the lubricating oil in a circulating system of a turbine due to rapid agitation, but sometimes, for no apparent reason, the oil begins to foam to the extent that the reservoir will overflow. So long as the mean desired bearing temperature is maintained, or if the bearing temperature is even slightly higher, there is nothing to be alarmed at over this excess foaming. Such excess foaming usually has no connection with permanent emulsions, but is caused by air getting to the system through leaks in the pump suction or being forced into the bearings by the ventilating system of an electric generator. When this air is agitated with oil, minute bubbles are formed which hold together to the extent that the volume increases until it overflows the reservoir. On many installations the drain lines are too high above the oil level in the reservoir so that the fall of oil is sufficient to cause a foam that rapidly accumulates. In either case when the air is released it carries with it some oil vapor, which increases the evaporation and the rate of deterioration of the lubricating properties in the oil. The main reasons for excessive foaming, however, are and mostly occur when additional oil is added to the oil in the circulating system, *while the turbine is running.* At the moment when such oil is added there is a large difference in the temperature of the oil in the system and the temperature of the oil added. This difference in the temperature causes unusual agitation in the oil due to reaction caused by air getting in the system when oil is added and when fluids of two temperatures are rapidly mixed. In order to avoid this it will be necessary to add oil in very small quantities at a time and the oil added must be of same temperature as the oil in the system, *i.e.*, about 4 degrees-5 degrees C. below the bearing temperature.

Usually when a turbine is started after the plant has been laying idle for some weeks, with the oil having been standing still in the

circulating system, the water cooling system is set to function notwithstanding the fact that the temperature of the oil in the system is low enough to run it through several times. This should not be done until the temperature of the oil has reached the normal temperature desired, otherwise in most instances there will be excess foaming.

In cases where a new turbine is started, after the required amount of oil for the circulating system supplied, the turbine must not be run more than 280 working hours with this oil. Then the turbine should be stopped, all the oil drained, and a fresh supply of oil supplied. On this fresh supply the turbine may be run up to 15,000 working hours, if the oil supplied is properly manufactured and prepared strictly according to the following specifications:

General Specifications of turbine oil referred to in this discussion are:

Specific gravity	0.890—0.905	
Viscosity Engler	2.50 —2.75 at 50° C.	
Flash point	190°C.	(Brenken open cup)
Color	10.0	(by Robinson)

Emulsion test—oil must readily separate from water at 40 degrees C. and after settling 30 minutes at this temperature the oil must show a fine line of separation between the oil and the clear water. The oil should be manufactured preferably from paraffine base crudes. It should be extra refined and doubly filtered and acid treated, but the result of such treatment must not leave any "Sulpho Compound" nor fatty acids in the body of the oil.

All turbine oils, while in the process of manufacture, and while stored should be stored in galvanized steel drums. Also in any laboratory tests oil should never be exposed to sun rays as the ultra-violet rays of the sun tend to cause oils to lose their power to resist decomposition of pure mineral lubricating properties, which in turn cause oils to have less resistance to emulsification.

All laboratory tests made should be carried out according to prescribed methods of U. S. Bureau of Standards, with the exception of the temperature values for settling in emulsion test.

DISCUSSION OF THE PAPER—"STEAM TURBINE LUBRICATION."

By Lieut. C. A. Griffiths, U.S.N., Member.

It has been known for a number of years that straight mineral oils are the best lubricants obtainable for turbine lubrication because of the better resistance they offer to oxidation and tight emulsification. While not so saying, it is presumed Mr. Riis' article deals exclusively with straight mineral oils.

Oil in a turbine lubrication system may come in contact with water from leakage or steam condensate and should it form an emulsion with this water the lubrication will be destroyed and clogged oil lines may result. Emulsification in a mineral lubricating oil is due to oxidation and the presence of "sulphur compounds." Sulphur and its compounds occur in natural crude. The oil may, during refining, also be treated with sulphuric acid and unless the sulphur-compounds from the original crude and after the acid treatment, during refining, are removed, emulsification will take place.

The water with which the oil in a turbine lubricating system may come in contact will generally be in one of the following conditions:

(1) Warm distilled water.
(2) Salt water.
(3) Water treated with boiler compound.
(4) Boiling distilled water.

An oil may not emulsify with warm distilled water yet, when mixed with alkaline water, a strong emulsion may take place. To be suitable for turbine lubrication, it must separate out from water in any of these conditions.

A new oil having a relatively large percentage of sulphur compounds may possibly readily separate out from an emulsion. This same oil during use under high temperatures and pressures will oxidize, and the presence of the sulpho-compounds will hasten the oxidization. Emulsion is therefore dependent on the degree of

oxidization and after a short period of use this oil will not separate out from an emulsion.

Naval minimum requirements guard against the possibility of a poorly refined oil being used in turbine lubrication. To preclude the acceptance of an oil which, by the presence of sulpho-compounds, will oxidize rapidly the naval requirements cause the rejection of an oil having any Free Sulphur, Corrosive Sulphur Compounds or SO_3 as Sulphonates, and more than a definite percentage of unsaturated compounds. These requirements likewise demand that the oil not emulsify with any of the above mentioned water conditions.

For example, in the case of the grade of oil mentioned by Mr. Riis as being suitable for the turbine on which he conducted his investigation, the naval requirements in the above respects are as follows:

(1) No fixed or fatty oils are permitted.

(2) There must be no free sulphur nor corrosive compounds.

(3) There must be no SO_3 as sulphonates.

(4) Unsaturated compounds must not exceed 6 per cent.

(5) In the emulsion test the oil in an emulsion at 130° F. must settle out:

(*a*) From distilled water in 30 minutes.

(*b*) From 1 per cent salt solution in 30 minutes.

(*c*) From normal NaOH in 30 minutes.

(*d*) From boiling distilled water in 30 minutes.

(6) It must not emulsify with steam.

All oils used in the Naval service are tested for acceptability at the Engineering Experiment Station, Annapolis. The usual chemical and physical tests to ensure conformity of the oil with the prescribed minimum requirements are first given. After the oil has satisfactorily met these requirements, it is run in an "endurance testing machine," which consists of a single motor driven journal operating at a constant speed of 2000 R.P.M., for a period of 100 continuous hours. During this time the oil is supplied at constant pressure to the journal by a gear driven pump. At the end of the test the oil is removed and its characteristics again obtained and compared to those determined for the new oil.

During the test on the endurance machine conditions are similar to those obtaining in a pure turbine bearing except that water is not present. This test resembles, as closely as possible in a laboratory, conditions existing in actual service, and the oil under test is subjected to the same elemental forces as is the oil in an actual turbine bearing. As all possible variables in the mechanical construction and operation of the machine have been eliminated, what differences noted between the used oil and the new oil are attributable only to the changes which occur in the composition of the oil itself. Oxidation of the oil is the first apparent change and with increased oxidation there naturally occurs an increase in the tendency of the oil to emulsify.

This increase in emulsibility of the oil has been verified in a number of experiments, some of the results of which for the grade of the oil being discussed are as follows:

UNUSED OIL, TIME TO SEPARATE OUT IN MINUTES.

	SAMPLE.					
	A	B	C	D	E	F
Distilled water	5	8	4	5	7	4
1 per cent Salt	2	3	2	1	3	1
Normal NaOH	1	15	7	3	3	1
Boiling distilled water	6	20	8	8	10	8

USED OIL, TIME TO SEPARATE OUT IN MINUTES.

	SAMPLE.					
Distilled water	35	29	7	19	40	15
1 per cent Salt	15	24	15	15	30%*	45
Normal NaOH	25	30	15	6	35	40
Boiling distilled water	60	37	25	21	60	50

PERCENTAGE OF INCREASE IN TIME TO SEPARATE OUT.

	SAMPLE.					
	A	B	C	D	E	F
Distilled water	600	273	75	280	472	275
1 per cent Salt	650	700	650	1400	--	4400
Normal NaOH	2400	100	114	100	1057	3900
Boiling distilled water	900	85	212	152	500	525

* Per cent emulsion present at end of one hour.

While it is conclusively known that the emulsibility of an oil increases with working, the relation between the factors tested which causes this increase in emulsibility cannot be definitely stated other than to say that the more carefully the oil is refined, the slower will be the rate of oxidation. There is no sequence in the above table nor visible any connection between the different periods of time. Obviously then something either occurs to the oils during refining, or are inherently properties of the crudes, which cause this variation and which research has not yet discovered.

Problems such as Mr. Riis has encountered lead to the goal of knowledge toward which all research strives. Mr. Riis has discovered that different oils will emulsify to different degrees under identical conditions of operation; that the same oil will emulsify to different degrees under different conditions of operations. The Engineering Experiment Station has realized this condition for some time and is continually working toward the connection between the degree of emulsification and elimination of the causes.

The subject of the proper turbine lubricant does not, as is inferred from the article, end with the non-emulsifying property of the oil. There are many other characteristics each of which plays a definite part in the lubrication problem and the oxidation of the oil. Among other requirements, the oil must not oxidize to the extent that serious sludge formations are encountered. It must show a neutral reaction. It must have low carbon forming propensities, and the evaporation loss must be a minimum.

In performing the standard chemical and physical tests the Navy is guided by the procedure covered by the Federal Specification Board which is in turn based on the results obtained by the Bureau of Standards. Where these test methods fall short, the Navy has developed its own, as in the case of the endurance testing machine. While the differences between paraffin and napthene base crudes are known, still the Navy does not recognize the former as being superior to the latter in lubricating qualities, as would be necessary were the requirements proposed by Mr. Riis adopted.

NOTES.

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NORTH GERMAN LLOYD LINER *BREMEN*.

Description of Machinery Installation*.

By W. Koch, Bremen. Translated by E. C. Magdeburger.

Successful designing, building and operation of machinery installations of large express liners require exceptional performance and the closest of co-operation on the part of both the building yard and the operating organization. This can only be done when both organizations are willing to lighten each other's burden of responsibility and have complete confidence in the mutual ability and willingness to surmount all difficulties. This important consideration has been guiding the builders and owners of the S. S. *Bremen* throughout the period of creation of its machinery installation.

CONSIDERATIONS ON THE CHOICE OF DRIVE.

The following requirements were communicated to the builders in October, 1926, for the purpose of preparing a preliminary design of the machinery installation of the *Bremen:*

Only geared turbines with oil-fired boilers and highly preheated feed-water should receive serious consideration. A certain portion of auxiliaries, such as ice machines and steering engine should therefore be steam driven; the electric light and power requirements, however, had to be supplied by Diesel driven generators. A speed of only 24 knots was considered at that time and the required propelling power was to be distributed on two shafts.

In November of that year the North German Lloyd increased the required service speed to 26¼ knots. From then on the twin screw arrangement had to be abandoned and only a quadruple screw installation was considered. The owners also soon decided in favor of water tube boilers although the same number of Scotch boilers of equal capacity could be installed in the same fire room space.

This decision resulted in a large reserve of power, since the evaporation load per unit of heating surface of the water tube boilers had to be kept much lower in future operation, even for a considerably larger steam consumption, than it would be possible with Scotch boilers of equal total heating surface. Besides with water tube boilers higher steam pressure could be adopted which would enhance the overall efficiency of the installation. The height of the superstructure could also be reduced so that in addition to the above advantages reduction of cost of the ship's hull had to be expected also.

In the early part of December the first design of the ship was completed by the Weser yard; this design, however, did not become the basis of the final machinery installation of the ship which soon thereafter was contracted for. The final design was approved only in January, 1927, soon after the merger of the Vulcan-Werft and A. G. "Weser" into the present "Deschimag" organization (Deutsche Schiff-und Maschinenbau- A. G.).

PARTICULARS OF DESIGN.

Immediately upon signing of the contract it was decided to begin with a design of an experimental water tube boiler, which would have to be officially approved and subjected to a series of tests on shore. The selection of the type of boiler was influenced by the experience of the Vulcan yard in the building of water tube boilers after the war and the operating experiences and extensive tests of the North German Lloyd on their small coastwise

* Published in Zeitschrift V.D.I. 24 May 1930, p. 693.

steamer *Roland* equipped with water tube boilers. An improved type of boiler consisting of one upper steam drum with as large as possible water surface and two lower drums, the superheater coils being arranged between the tube bundles, was thus arrived at. The most satisfactory size of heating surface of each boiler was determined through the above mentioned experiments and thereby the number of boilers and the division of the total required capacity into four fire-rooms decided on—each of these to furnish steam for its own turbine groups of the four shaft installation. A steam pressure of 23 atmospheres (327 pounds per square inch) was chosen.

To permit passage of various social halls on the upper decks through the center of the ship, the uptakes could not be arranged amidships as customary, but had to be carried upward divided on both sides. The uptakes of each of the two groups of boilers were combined quite high in the superstructure into one comparatively short smoke pipe.

Considerable discussion preceded the decision on the method of supplying combustion air to the fire rooms. This question was made particularly difficult for the owners since the builders of the sister ships *Bremen* and *Europa* were divided in this respect, while one desired to install the "closed fire-room" system the other advocated "Howden heated forced draft". Owners and builders were endeavoring to appraise the advantages and disadvantages of both these systems as objectively as possible. An inspection trip by all concerned was made to investigate in addition also the possible methods of fire-room ventilation. The "closed fire-room" installation appeared especially suited to obtain frequent change of air in the fireroom. It was decided therefore to agree with the proposal of the "Deschimag" and install this system, although it is only seldom used in merchant marine. The blowers were to have sufficient reserve capacity and their wheels made strong enough to supply increased quantities of air to the fire rooms during hot weather through especially provided regulated openings to the spaces between the smoke pipe and its housing.

Since two sisterships were being built the owners quite naturally desired their machinery installations to be as similar as possible, since this would simplify the future operation of the two vessels appreciably. This desire, however, especially as far as the design of the main turbines was concerned could not be heeded.

The question of materials received maximum consideration as had to be expected. For a long time for instance the "Izett" steel, manufactured by Krupps was considered for the drums of the water tube boilers. Exhaustive tests were conducted by both yards, the owners and the boiler inspection bureau on this material and showed otherwise excellent qualities. However "Izett" steel at that time was not uniform enough and could not be produced in quantities sufficient for the requirements of both ships, necessitated by the short terms of delivery. The selection therefore fell upon boiler steel of 47 to 59 kilograms per square millimeter (67,000—77,000 pounds per square inch) tensile strength with an elongation of at least 20 per cent for the shell of the upper drum, while a steel of 41 to 48 kilograms per square millimeter (58,000—68,000 pounds per square inch) with an elongation of 25-20 per cent was specified for the rest of the boiler.

The proper material for the extensive insulation was also selected only after exhaustive tests—for the piping the 85 per cent Original-Newalls-Magnesia was specified while for covering of boilers spunn-glass in the shape of mats was to be used. Much consideration was required to agree on the type of flange joints to be used on main and auxiliary steam piping. Not only the material of boiler tubes but the method of their rolling into the drums were also exhaustively tested.

As a rather important circumstance, it must also be mentioned that no chemical treatment of feed water was to be used since only distilled water would be utilized for feeding the boilers. Fresh water taken from shore for boiler feed purposes was not only to be distilled as needed but provisions made to carry large stores of such distilled water on board.

While the above shows the measures taken to assure reliability of boiler installation, similar efforts were made to make the main turbines themselves of equal reliability. Here also the selection of materials was made with great care. For strength reasons and to better meet thermal stresses the housings of the turbines were made of cast steel, which is not frequently done. As turbine blading material were chosen Monel metal and V5M steel and only after considerable experimentation it was decided to use the so-called Admiro-Bronze for making labyrinth packing rings of the main turbines.

GENERAL ARRANGEMENT OF MACHINERY.

Table I shows the general arrangement of machinery and boilers in various compartments. Eight water-tight compartments between frames 67 and 234 house the entire machinery installation. The four fire rooms housing 20 water tube boilers with an ice machine and pump room in the center between the forward two and after two groups of boilers occupy the space between the frames 123 and 234. The forward engine room housing the propelling turbines of the wing shafts lies between frames 104 and 123, while the after engine room with the turbines of the center shafts extends from frames 82 to 104. Further aft, between frames 67 and 82 is located the auxiliary engine room with Diesel generator units for electric light and power. Following this four shaft tunnels extend aft up to the respective stern tubes at frames 22 and 49.

In these rooms a large number of various auxiliary apparatus is installed, the location and name of each is to be found by means of plate I and the legend explaining the letter designation of each. The importance of these auxiliaries to the operation of the whole plant will be discussed briefly later.

MAIN TURBINE INSTALLATION.

Four identical in size but independently operating turbine units propel the ship and each drives its own shaft through a gear drive. The installation is designed for surface condensation and boiler pressure of 23 atmospheres (327 pounds per square inch) and superheated steam of 350 degrees C. (662 degrees F.) at the turbine. Each turbine unit consists of a high pressure ahead turbine, intermediate pressure ahead turbine, and low pressure ahead turbine as well as a reduction gear drive—Figure 1. In the housing of the intermediate ahead turbine is the high pressure astern turbine while the low pressure astern turbine is in the housing of low pressure ahead turbine. The reversing power is about 65 per cent of the full power ahead.

The housings of all turbines are made of cast steel. To inspect a part of the blading without dismantling the housing is provided with suitably located peek-holes. Man holes are also provided to permit turning over of turbines for inspection of blading and shrouding. All turbine housings are insulated in the most careful manner with Newalls-Magnesia of 85 per cent. In such places where insulation may be required to be taken off periodically separate asbestos mats are used. The turbine housings are so connected to the foundation girders as not to interfere with their free expansion under

heat. Under the turbine supports are provided cast brass plates to permit sliding. Special cast steel nozzle boxes serve to admit boiler steam to high pressure ahead and astern turbines so that for partial loads no one-sided heating and distortion can take place.

The outside glands are so designed that they may be removed without lifting the turbine casing. Connections for steam seal and drainage are provided. The high and intermediate pressure turbines have dummy pistons and labyrinth packing. The turbine disks are made of steel of especially high tensile strength and elongation. The turbine rotors were dynamically balanced upon completion of blading.

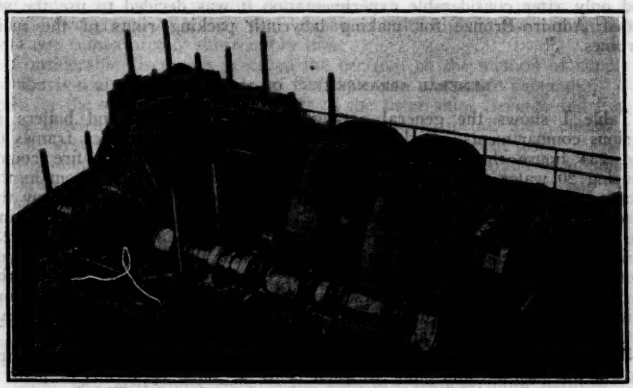


FIG. 1.—MAIN TURBINE UNIT ON THE ASSEMBLY FLOOR.

The high speed turbines are connected with their respective pinions by means of axially sliding double jaw couplings so that should a turbine be disabled it can be disconnected from its pinion without dismantling any larger parts. The jaws of the couplings are made of such a material that they will not seize.

On the free end of each turbine is installed a shut-off governor, which operates a governor valve in the main steam line to the turbines when the speed of the turbine exceeds by about 10 per cent its maximum designed R.P.M. The governor valve comes also into action when the supply of lubricating oil to the main bearings and the gears for some reason or other be discontinued or the turbine rotor due to some defect will develop excessive pressure on its thrust bearing by sliding axially. The pressure oil to operate these safety devices is taken from the lubricating pressure line of the turbine.

The superheated steam coming from the boilers passes first through a water separator, then through the governor valve to the main trottle valve for ahead and astern operation. Before reaching the inlet nozzle groups of the impulse wheels (Curtis type) of the H.P. ahead or H.P. astern turbine the steam passes through a steam strainer. Under overload conditions the major portions of the live steam by-passes the nozzle groups through an overload valve direct into the impulse wheel chamber of H.P. ahead turbine. The main manoeuvering valves are balanced and are manipulated by the operator through one ahead and one astern handwheel which are so interlocked that a simultaneous opening of both valves is impossible.

From the H.P. turbine the steam exhausts into the I.P. turbine through a transfer pipe while two pipes serve to convey I.P. exhaust steam to L.P. turbine, from where it finally reaches the condenser through a large exhaust elbow. Transfer pipes between the ahead turbines are corrugated. Provision is made to run any two turbines of the unit when one of the three turbines will have to be disconnected for some reason or other. The axial location of the turbine rotor and the pinion may be conveniently inspected any time while running; but suitable measurements of any dislocation of the shafts in the vertical plane may also be made. To assure good drainage of the turbines pockets are provided in their casings at the points of collection of condensate water, especially in the bottom of the low pressure unit, and all the pockets of each unit joined by means of drain pipes to a common receiver from which the condensate is pumped by specially provided pumps. For warming up the turbines there is a small by-pass pipe through which live steam may be admitted to the H.P. turbine. Traveling cranes and trolley hoists are provided over the turbines to permit speedy overhaul.

REDUCTION GEARS, SHAFTS, PROPELLERS.

The turbine speed at normal ship speed is about 1800 R.P.M. which is reduced by means of a single reduction gear to about 180 R.P.M. of the propeller shafts. Each reduction gear consists of a double rim gear wheel, which is engaged by three pinions of the H.P., I.P. and L.P. turbines—Figure 1. The gear shaft is coupled to the thrust shaft and this in turn is coupled to the propeller shaft. The pinions of the I.P. and L.P. turbines are located in the horizontal plane of the shaft centerline, whereas the H.P. pinion is directly above the gear wheel. Gears are of the customary herring-bone type and were cut by special machinery. The accuracy of the teeth was checked in the most careful manner with special instruments. The gear housing is made of cast iron and is so subdivided that each pinion may be removed by itself and the teeth of the large wheel may be inspected any time without lifting the upper half of the housing. Each turbine unit has on the free end of the L.P. pinion shaft an electrically operated turning device which is used also while warming up the unit.

The pinion shafts are made of alloy steel and are axially movable so that they can adjust themselves to the position of the gear wheel. Power is transmitted through two separate pinions to the two rims of the gear wheel. The gear wheels are of cast iron and made separate, although held together by substantial bolts. The gear wheels are held by key on the tapered shaft and have shrunk-on rims of suitable alloy steel. The main bearing shells of the pinion shaft are of forged steel, while those for the gear wheel shaft are of cast steel, and all white metal lined. Lubrication of the reduction gear is connected to the main pressure lubrication system. The pinions are provided with special oil distributing manifolds and these are so arranged that they are easily accessible for inspection.

The line shafting has been considerably increased in diameter as compared with the requirements of the classification societies in order to take care of torsional vibration. The spring bearings are of the ring type, the housing being provided with hollow spaces for water cooling. The line shafting is connected to the propeller shaft by means of a two piece muff coupling. The propeller shafts have bronze bushings shrunk on them and between the forward and after bushings received a rubber covering.

The thrust bearings are on the shaft coupled to the after end of gear wheel shaft. They are of the single collar type—Figure 2—and the heavy flange of the lower half of their housing is bolted to the foundation and

the ship's hull. The thrust is transmitted to the foundation through strong forged steel plates riveted to the foundation. The stern tubes were made of two parts due to their great length and fitted with forward and after bearings of lignum vitae. They have on their forward end a flange by means of which they are fastened to the bulkhead; the after end of the stern tube is fitted with thread and held by a special hammer nut in the strut.

The propellers are four-bladed and cast in one piece from special bronze. They are of 5.0 meters (16.4 feet) in diameter, of 5.2 meters (17.05 feet) pitch and weigh about 17 tons.

CONDENSER INSTALLATION.

Exhaust steam leaving each of the four L.P. turbines passes through large elbows equipped with expansion piece into surface condensers with a single water pass—Figure 3. The cooling surface of each condenser is 1760 square

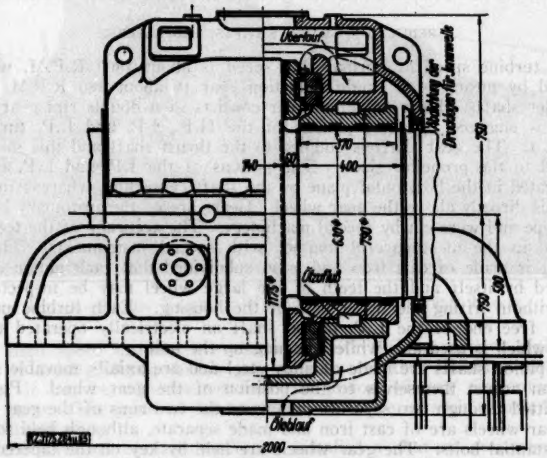


FIG. 2.—MAIN THRUST BLOCK.

Überlauf—Overflow; ölzufluss—Oil Supply; ölablauf—Oil drain.
Abdichtung der Drucklager für Innenwelle—packing of inside shaft.

meters (19,000 square feet). The two condensers for the outboard turbine units are arranged athwartship while those for the inboard units are located lengthwise in the ship—Plate I—in each case, however, so that the tubes may be easily removed for renewal without disturbing any of the larger ship or engine parts. The condensers are supported by means of feet on the side in such a manner that should large scale tension or compression forces develop free elongation may take place. The condenser casing is of forged steel, the tube sheets as well as the baffle plates are of Muntz-metal. In the lower part of the steam space a number of tubes is separated and serves primarily for cooling of air. Should excessive cooling of the condensate take place it may be helped by regulating the amount of circulating

water in the upper and lower halves of the condenser. In addition to the main condenser a small auxiliary condenser is provided for port operation.

To drain the main condensers and the main propelling turbines two turbine driven condensate centrifugal pumps are provided—Figure 4. These units are of vertical construction and both pump wheels are in one housing one above the other, held by a nut on the common shaft. Their casings are, however, separate—the drain pump of the turbines is in the lower casing

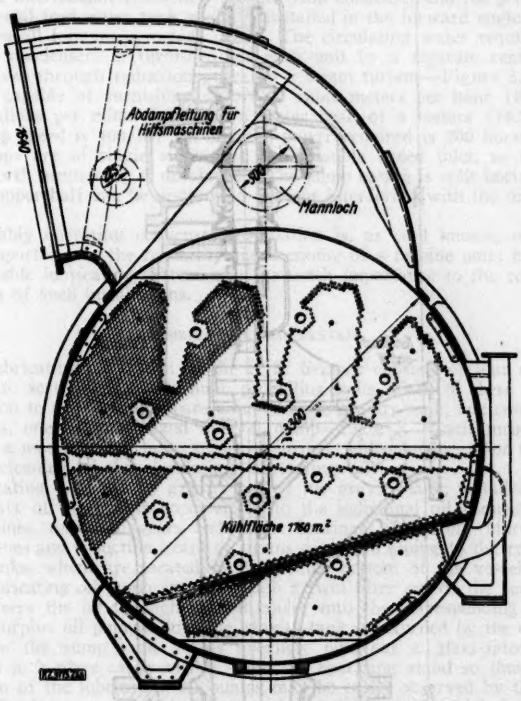


FIG. 3.—MAIN CONDENSER.

Abdampfleitung für Hilfsmaschinen—exhaust of auxiliaries.
Manloch—manhole; Kühlfläche—cooling surface.

while the condensate from the condenser is handled by the upper pump. The condensate outlet is carried up to the same level with the inlet. The pump shaft is direct coupled to the turbine shaft by means of a flange coupling. Below this coupling the pump shaft enters the casing through a water lubricated gland which is the only stuffing box on this shaft. A three stage impulse turbine drives the pumps—its casing is split axially and separate dismountable housings are provided for the turbine and its two bearings.

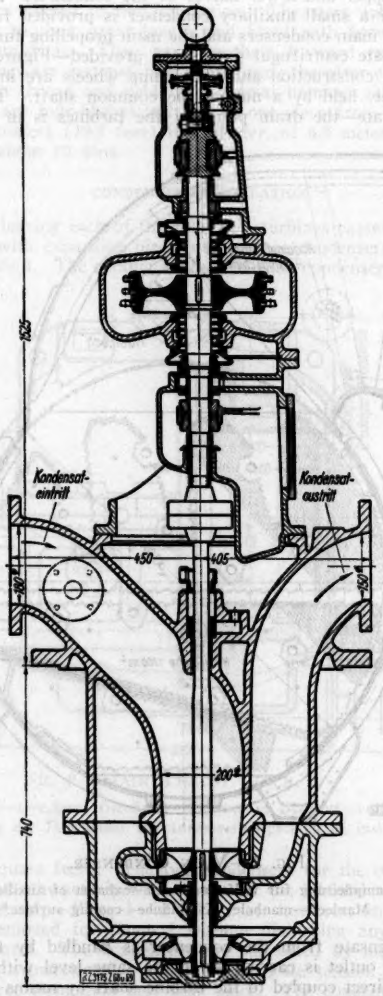


FIG. 4.—CONDENSATE AND DRAIN PUMP SET.

Kondensateintritt—condensate inlet.
Kondensataustritt—condensate discharge.

To remove the air out of the four condensers four steam jet air pumps are provided—Table I. Each of these units consists of two two-stage steam jet air pumps of which one serves as a spare and operates in connection with an intermediate condenser and a preheater; these are built into the condensate discharge line and are passed by the condensate one after the other. The air pump units are bolted on to the horizontally disposed intermediate condenser and preheater in the condensate line. A water seal drains the intermediate condenser into the main condenser and the preheaters into a special feed water tank which is installed in the forward engine room and serves all four main turbine units. The circulating water required for the main condensers is furnished for each unit by a separate centrifugal pump driven through reduction gears by a steam turbine—Figure 5. Each pump is capable of furnishing 6000-7000 cubic meters per hour (26,700—30,800 gallons per minute) against a water head of 6 meters (19.7 feet). The pump speed is 400 R.P.M. and the power required is 200 horsepower. The pumps are of single stage type with double ended inlet, so that no thrust worth mentioning is developed. The pump casing is split horizontally and the upper half can be dismantled without interfering with the discharge piping.

A reliably operating condenser installation is, as well known, of paramount importance to the reliability and economy of a turbine unit; however, a dependable lubricating system is of as much importance to the continued operation of such installations.

LUBRICATING OIL SYSTEM.

The lubricating oil system of the S. S. *Bremen* consists of four separate systems to serve each of the main propelling units. Each of these systems in addition to the required spare units has one gravity tank, one sump tank, oil filters, one oil cooler and one oil pump—Plate I. Each turbine unit has also a motor driven lubricating oil purifier with equipment for the continuous cleaning of the circulating lubricating oils.

Lubricating oil flows by gravity out of the gravity tank, installed in the upper part of the engine room shaft, to the individual oil connections on the turbines, reduction gears and thrust bearings. Used oil returns from the turbines and reduction gears by means of return piping to the respective sump tanks, which are located in the double bottom of the vessel. Each main lubricating oil pump sucks through a twin filter out of the sump tank and delivers the oil through the oil cooler into the corresponding gravity tank. Surplus oil pumped into the gravity tank is returned by the overflow pipe into the sump tank. This overflow pipe has a glass-sided fitting installed in a place easily visible from the operating stand so that correct operation of the lubricating oil pumps may be easily observed by the operator. Furthermore, should irregularities develop in the lubricating oil circulating system an electric alarm contact signal attracts the attention of the operator. The more important bearings of the turbines and reduction gears are fitted with thermometer-wells so that they can be easily watched while larger lubricating oil drain pipes have glass gauges to show circulation of oil.

The main lubricating oil pumps—Figure 6—are of the vertical centrifugal type and are direct turbine driven, developing their rated capacity at 3200 R.P.M. Since the reliable performance of the whole turbine plant depends so much on perfect operation of these pumps features of their design were chosen with particular care to insure reliability of operation.

Figure 6 shows the section through this pump unit with axial inlet and exit to impeller. The shape of the vanes in the bronze pump wheel was

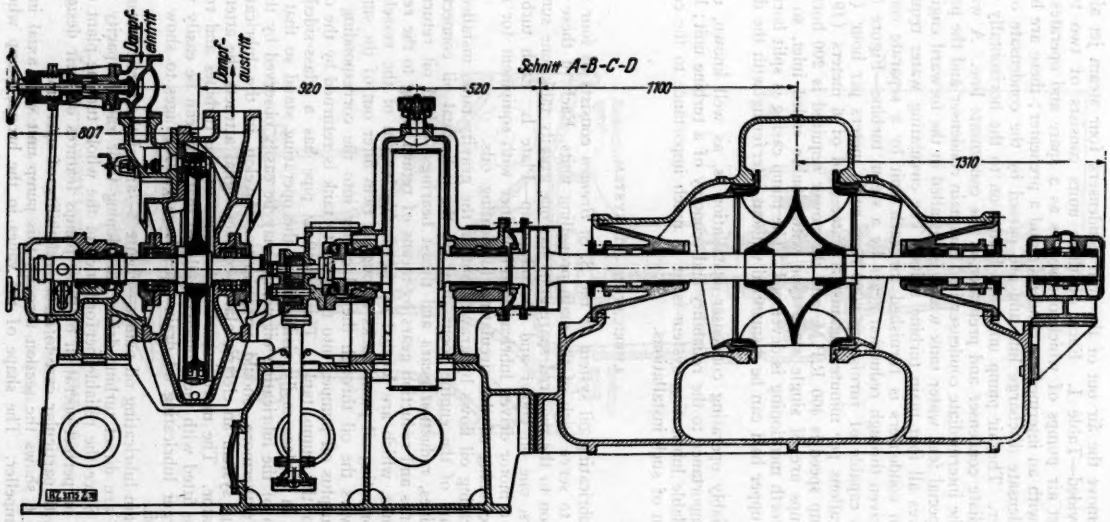


FIG. 5.—MAIN CIRCULATING WATER PUMP

Dampfeintritt—steam inlet; Dampfaustritt—steam exhaust.

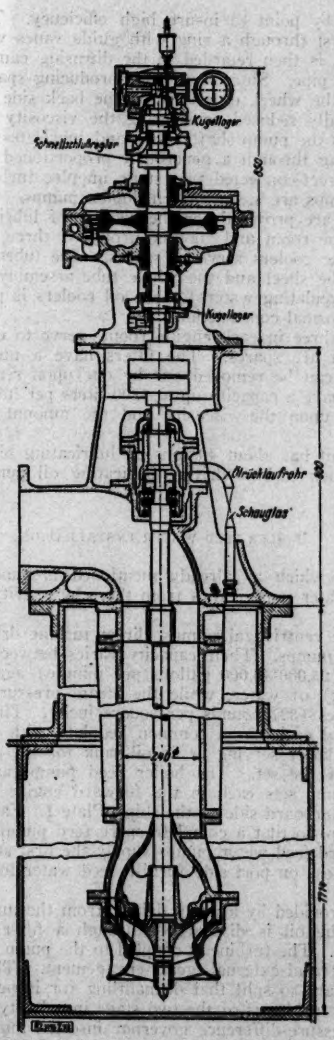


FIG. 6.—MAIN LUBRICATING OIL PUMP.

Kugellager—ball bearing; Schnellschlussregler—governor.
Ölrücklaufrohr—oil drain pipe; Schauglass—gauge glass.

determined point by point to insure high efficiency. The oil leaving the impeller passes first through a ring with guide vanes where it changes its direction to axial, is then retarded in the diffusing canals and reaches the vertical discharge pipe. Since large eddy-producing spaces do not exist on the periphery of the wheel, especially on the back side of it the hydraulic losses are materially reduced considering the viscosity of the oil handled. The oil surrounds the pump shaft above and lubricates all of its bearings, leaving the housing through a generously proportioned elbow. The pump is driven by a direct-connected two stage impulse turbine. The auxiliary lubricating oil pumps are steam driven duplex pumps.

Six oil coolers are provided for the cooling of lubricating oil—three in the forward engine room and another group of three in the after engine room. Two of the coolers serve as spares. The tubes of the coolers are rolled into the tube sheet and the whole tube assembly can be withdrawn upwards. The circulating water for the oil coolers is provided by vertical steam pumps of normal construction.

Six oil filters—three in each engine room—serve to clean the oil. Here also two of them are spares. The filters have a number of individual filter bags which can be removed one by one upon removal of the cover. The oil purifiers have a capacity up to 2800 liters per hour (735 gallons per hour) depending upon the viscosity and the amount of foreign matter in oil.

Each turbine unit has about 40 tons of lubricating oil in circulation and the capacity of each turbine driven lubricating oil pump is 270 tons per hour.

BOILER FEED WATER INSTALLATION.

The condensate which as already mentioned is pumped by condensate pumps into feed water tanks, flows from there by gravity to the feed water pumps.

Eight two stage centrifugal pumps, direct turbine driven serve as main boiler feed water pumps. Their capacity varies between 95 and 175 cubic meters per hour (25,000-46,000 gallons per minute) against a head of 320 meters (1050 feet) of water, while the steam pressure in the boilers is only 23 atmospheres (327 pounds per square inch). The pumps are of the horizontal type and each has a common bedplate with the turbine driving it—Figure 7—the latter serving as an oil tank for the pressure lubrication of the bearings of the set. The boiler feed pumps are installed in two groups of four pump sets each in the forward engine room, one on port and the other on starboard side of the ship—Plate I. Each group of boilers is fed by one pump, so that a complete spare feed pump is therefore available. The starboard feed water pumps supply the first and the third groups of boilers while those on port side furnish feed water to second and fourth fire-rooms.

Lubrication is provided by a pump driven from the turbine shaft through a worm gear. The oil is discharged through a filter and cooler to all bearings of the set. The turbine is coupled to the pump through a coupling with a double internal-external gear engagement. The casings of the turbine and pump are so split that dismantling for inspection and overhaul is quite easy. The turbine is of the two stage impulse type and is regulated by a hydraulic pressure-difference governor inserted into the steam supply pipe. This governor automatically reduces the R.P.M. of the turbine by acting on its throttle valve, so that the pressure in the boiler feed pipe always remains higher than boiler pressure by a safe margin.

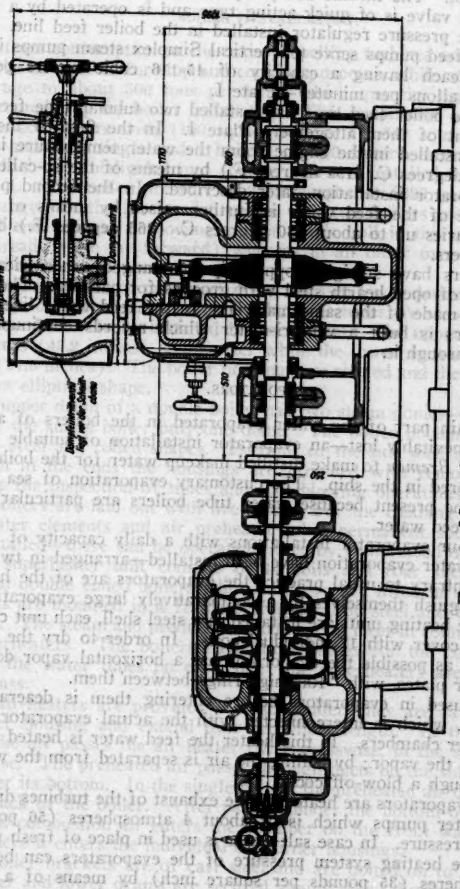


FIG. 7.—MAIN FEED WATER PUMP.
Dampfeintritt—steam inlet; Dampfaustritt—steam exhaust.
Dampfeintrittventil liegt vor der Schnittebene—
steam inlet valve is before the section plane.

The pump has two stages with double-suction impellers so that no axial thrust can develop. For maintaining the proper axial position of the impellers a small single collar thrust bearing is installed on the free end of the pump shaft. A revolving hand, driven from the pump shaft end permits the operator on the upper platform to see whether the pump sets below run or not. The turbine and pump wheels are dynamically balanced. The steam inlet valve is of quick acting type and is operated by a Hannemann membrane pressure regulator installed in the boiler feed line.

As auxiliary feed pumps serve ten vertical Simplex steam pumps installed in fire rooms, each having a capacity of 45-116 cubic meters per hour (11,600-30,600 gallons per minute)—Plate I.

In each of the boiler feed lines are installed two tubular type feed water heaters—or eight of them altogether—Plate I. In the first of these feed water heaters installed in the engine room the water temperature is raised up to about 90 degrees C. (194 degrees F.) by means of the so-called vapor from the evaporator installation, later described. In the second preheater the temperature of the feed water is further raised by means of exhaust from the auxilaries up to about 130 degrees C. (266 degrees F.) before it reaches the boilers.

The preheaters have straight copper pipes, arranged concentrically—the tube plates are of open hearth steel with grooves for packing. The casings are parted and made of the same material. In the feed water line of each group of boilers is built a venturi-meter which records continuously the flow of water through it.

EVAPORATORS.

Since a certain part of the water evaporated in the boilers of a marine installation is inevitably lost—an evaporator installation of suitable size was provided on the *Bremen* to make distilled makeup water for the boilers from fresh water stored in the ship. The customary evaporation of sea water is not used for the present because water tube boilers are particularly sensitive to salt in feed water.

A total of four evaporator installations with a daily capacity of 75 to 85 tons of fresh water evaporation have been installed—arranged in two groups —Plate I. Contrary to usual practice the evaporators are of the horizontal type and distinguish themselves by a comparatively large evaporating area. Two removable heating units are installed in a steel shell, each unit consisting of a cast iron cover with 17 individual coils. In order to dry the resulting vapor as much as possible the evaporator has a horizontal vapor dome with removable filter plates, with "Raschig" rings between them.

The water used in evaporators before entering them is deaerated in a mixture-heater, which is interconnected with the actual evaporator with its steam and water chambers. In this heater the feed water is heated up to the temperature of the vapor; by boiling the air is separated from the water and blown off through a blow-off cock.

At sea the evaporators are heated by the exhaust of the turbines driving the circulating water pumps which is of about 4 atmospheres (56 pounds per square inch) pressure. In case salt water is used in place of fresh water for evaporation the heating system pressure of the evaporators can be reduced to 2.5 atmospheres (35 pounds per square inch) by means of a reducing valve provided in the inlet pipe.

The vapor from the evaporators is used for heating boiler feed water, as has been already mentioned. Special pumps installed in the firerooms supply the water to the evaporators—they discharge the raw water into a gravity

tank, from where it flows through a water level regulator on the evaporator first into the above mentioned mixture-heater. At sea therefore the vapor leaving the evaporators for the feed water heaters compensates for the unavoidable feed water losses, which, however, are caused not so much by the main propelling turbines as by the numerous smaller steam consumers.

BOILER INSTALLATION.

The steam consumption of the main propelling turbines and all the various auxiliaries as well as for ship and cooking purposes, while at sea amounts on an average to about 500 tons per hour, which quantity must be furnished uninterruptedly by the boiler installation. This installation consists of eleven double-ended boilers with 1100 square meters (11,840 square feet) heating surface and nine single ended boilers of 550 square meters (5920 square feet) which are subdivided into four groups and installed in four firerooms —Plate I. The boiler pressure is 23 atmospheres (327 pounds per square inch) gauge. In the foreward fireroom only two boilers are installed in the forward end (due to narrowing of the ship's hull in this compartment located already quite far forward) whereas in all other fire rooms there are three boilers in each row athwartships.

The boilers consist of one upper drum and two lower drums connected with two groups of curved tubes on each side, these tubes are so separated to permit a superheater to be installed between them. The upper drums of 1700 millimeters (43.2 inches) in diameter while the lower ones are of 850 millimeters (21.6 inches). The boiler housings are riveted and the riveted bottoms are of an elliptical shape.

Each upper drum of a double boiler has two steam domes—Plate II—while those of the single boiler only one dome. The boiler housings are made up of sheet steel and rolled shapes. Portions of the side wall may be removed and even in the ends the housings are provided with removable covers to facilitate the inspection of tubes and soot blowing operations. The fire box of the boilers are laid out with fire brick. For the cleaning of water tubes, superheater elements and air preheater tubes permanent soot blowers have been installed—these can be operated either by compressed or preheated air.

Each superheater unit consists of 14 coils and four such units are installed in double ended boilers while only two are fitted in single end units. The coils are mounted in a frame of forged alloy steel and the whole is removable. The steam and products of combustion meet in counterflow through the superheaters. The boilers have three passes for the combustion gases through the water tubes on each side; the superheaters are located in the second pass.

Two air-heaters are provided for each boiler, they are fastened direct to the boiler housing and extend over its whole length. In the double-ended boilers the air passes the air heater twice crosswise, in the single ones once lengthwise. The preheated air passes through ducts on the side of the boilers and under its bottom. In the single boilers the preheated air duct is arranged on the back of the boiler. After passing under the bottom of the boiler the preheated combustion air enters the chamber on the front wall of the boiler from where it enters the individual oil burner housings.

The boiler brackets are of cast steel and are bolted on to the foundation. The upper and lower drums are secured against axial displacement by anchor bolts fastened to respective bulkheads. All boilers have high lift safety valves of Kockburn design and are fitted out with other customary fittings. Boiler feed water is regulated by so-called Mumford regulators. The piping connecting the float housings of these regulators to the steam and water of the

upper drum has valves in it so the regulators can be overhauled with the boiler under steam, by simply closing these valves. Tangential steam dryers are built into the steam domes of the boilers to assure that only the least amount of water is carried into the superheaters.

The total heating area of the complete boiler installation amounts to about 17,050 square meters (183,500 square feet) of the superheaters—3875 square meters (41,700 square feet) and of the air heaters—8786 square meters (94,600 square feet).

The installation of boilers was not without considerable difficulties, since a double ended boiler weighs about 90 tons. The boilers were delivered to the side of the ship on a skidway, lifted up over the bulkheads by means of two yard cranes and two spreaders and lowered into the firerooms, where another skidway was laid, to move the boilers sideways to the designation with block and tackle. All of the 20 boilers were installed prior to launching.

OIL FIRING.

The total consumption of fuel oil for a round trip is about 6200 tons. The fuel oil is stored partly in double bottoms and partly in high bunkers and is taken over into special filling compartments, arranged one on each side. These filling compartments are accessible from the firing rooms and have outboard ports through which filling hoses may be brought in. All oil tanks are equipped with heating coils and the condensate of these coils drains into so-called observation tanks, through which flows continuously the raw water on the way to evaporators. These tanks have glass ports through which may be observed any possible leaks in the bunker oil heating coils.

The high bunkers are surrounded by drain canals which prevent any leakage oil from spreading over the top of the double bottom. These canals lead to oil wells from where special dirty-oil pumps deliver it to dirty-oil tanks.

From the bunkers the fuel oil is pumped into special day or settling tanks —Plate I—the pumping being done by four vertical simplex steam pumps of 120 tons per hour capacity. The suction and filling pipe systems of the fuel oil bunkers are so arranged that it is possible to pump over the contents of a tank in the bow of the ship into a tank astern and from a tank on one side of the ship to a tank on the other side and *vice versa*. Interconnections between filling and delivery pipe systems permit the oil pumps on board to deliver oil from the ship's bunkers or even to pump the bunkers out into lighters should emergency docking be necessary.

The fuel service pumps which take the oil from the day tanks and deliver it to the oil burners on the boilers are duplex steam pumps of 12 tons per hour capacity. In each fire room there are also installed two suction filters, two pressure filters and two fuel oil heaters.

The oil burners of Vulcan type—Plate II—consist of a frame with a hinged door. The distribution of air is done by guiding vanes and a sliding door and the air is given a whirling motion. The oil atomizer is encased in a central tube of the swinging door of the oil burner. This tube has on its outside end a forging with a quick closing valve. Special interlocking device prevents the lever holding the atomizer in the tube from opening before the quick closing valve is closed.

UPTAKES AND BLOWERS.

The products of combustion leave the boilers after passing through the air heaters and are led through ducts and uptakes to the smoke pipes—Figures 8-10. The forward and aft boiler groups have two vertical

uptakes—one on port and the other on starboard side of the ship—which combine into one only above the sun deck. The weight of the smoke pipes is taken up by a suspension system built into the deck structure in such a manner that both uptakes and smoke pipe may expand upward. In order to relieve the boiler housings from the weight of the uptakes these are suitably suspended on springs. The uptakes are divided into ducts leading from the individual boilers and only at the base of the smoke pipes these partitions disappear.

The firerooms are of the closed type, *i.e.*, the air required for combustion is delivered to the firerooms by means of special blowers. Communication between fire rooms and forward turbine room is possible through so called air locks, consisting of a small enclosure with double doors. Due to the pressure drop between smoke pipe and fire room the combustion air enters the air heaters on top of the boiler housings without any additional ducts; here it is preheated and reaches the oil burners through side ducts on the boiler housing.

The combustion air is furnished by eight turbine driven blowers. The turbines are of a two stage impulse type and drive a double-suction blower wheel through a reduction gear with single helical gears. An attached gear pump lubricates the reduction gear and the turbine bearings; it takes the oil out of a sump tank and delivers through a twin filter and an oil cooler direct to the places of consumption.

In port the firerooms are opened so as not to interfere so much with intercommunication and therefore three port boilers are equipped with a special electrically driven blower. These blowers deliver the required combustion air through ducts to the air heaters and from there to the oil burners as on other boilers.

DIESEL-ELECTRIC PLANT.

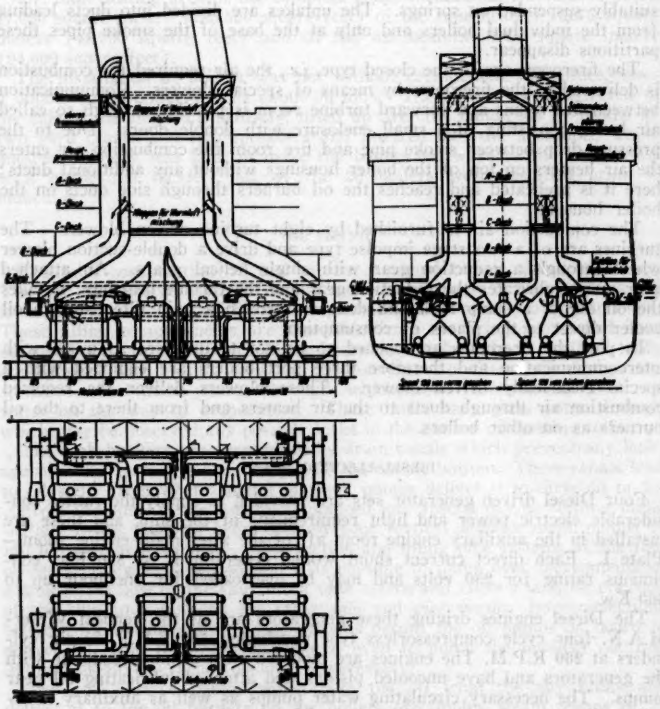
Four Diesel driven generator sets are provided to supply the rather considerable electric power and light requirements of the ship, and these are installed in the auxiliary engine room aft of the after main engine room—Plate I. Each direct current shunt wound generator is of 520 Kw. continuous rating for 230 volts and may be overloaded for one hour up to 560 Kw.

The Diesel engines driving these generators are of the vertical Weser-M.A.N. four cycle compressorless type developing 780 B.H.P. in six cylinders at 260 R.P.M. The engines are mounted on a common bedplate with the generators and have uncooled pistons and attached lubricating oil gear pumps. The necessary circulating water pumps as well as auxiliary lubricating oil pumps are motor driven centrifugal and gear type.

Each Diesel engine has eight vertical cast iron frames subjected to compression stresses only, while the tension stresses developed by internal combustion are taken up by tie rods, which hold the cylinder castings to the bedplate. The cylinders are equipped with liners of special cast iron, which can expand freely downward through a stuffing box. The cylinder heads are also of special cast iron with large core plugs, and are equipped with the usual inlet, exhaust, air starting, fuel injection and relief valves.

The valve housings and the seats of the inlet and exhaust valves are of special cast iron. The air starting valve is automatic, *i.e.*, it operates only when starting air is admitted to the starting air piping and it closes as soon as ignition occurs in the cylinder, thus fuel is also admitted when starting.

The pistons made of special cast iron are of the normal trunk type. The hardened and ground wrist pins are driven tight into the piston body and



Sonnendeck - Sun Deck.
Oberes Promenadendeck - Upper promenade deck.
Haupt " " - Main " "
Klappen für Warmluft mischung - Flapper valves for mixing warm air.
Kesselraum - Boiler room.
Gebläse für Hafenbetrieb - Blowers for port use.
Notausgang - Emergency exit.

FIGS. 8-10.—UPTAKES AND SMOKE PIPE.

are prevented from moving by tapered pins. The valve gear drive is on the flywheel side and helical tooth spur gears are used. The camshaft is in a casing filled with oil into which the cam noses dip to thus lubricate the contact surfaces. The valves are driven by push rods and all rolls and bolts as well as other contact surfaces under pressure are of hardened open hearth steel. The flywheel, arranged between the engine and the generator, carries teeth for turning the engine.

Individual pumps supply fuel oil to each cylinder, the amount being regulated by a by-pass valve, the opening of which is controlled by the governor to suit the load.

Each engine has on its free end a shut-off valve, for the starting air, combined with a pressure reducing valve, capable of reducing the air pressure from 75 atmospheres (1065 pounds per square inch) to about 25-30 atmospheres (350-425 pounds per square inch). The seamless drawn starting air tanks are made for an operating pressure of 75 atmospheres. Two motor driven auxiliary air compressors supply the starting air. They are of the two-stage type, equipped with the necessary intercoolers and aftercoolers. The lubricating oil draining from the Diesel engines is collected in a sump tank from where it is circulated again by the lubricating oil pumps. The lubricating oil is cleaned by a centrifugal purifier equipped with a heating tank.

The circulating water is furnished by three centrifugal pumps of which one serves as a spare. These pumps suck through strainer direct from sea and deliver the water through the oil coolers to the jackets of the engine cylinders.

The fuel oil is pumped from the storage tanks to the special day tanks by single-acting duplex steam plunger pumps. These duplex-pumps also pump the oil required by the oil fired kitchen ranges into a special service tank. The main switchboard is also in the auxiliary engine room.

EMERGENCY POWER AND LIGHT INSTALLATION.

Should the main source of light and power be disabled an emergency power and light installation may be started. It is located in a special room on the sun deck aft of the after smoke pipe. This installation consists not only of a power source quite independent of the main Diesel driven generator sets, but also of an emergency distribution system, which will furnish light to the vital places in the ship and operate a number of the more important motors in such an emergency.

Two vertical four cycle Diesel engines of the Weser-M.A.N. compressorless type are direct connected to generators and can furnish a total of 200 Kw. at 115 volts and 375 R.P.M. The fuel required for the operation of this installation is stored in four separate tanks on the uppermost deck and is sufficient for a four-day full-power operation of the installation. All the pumps and other apparatus required for the independent operation of the emergency power plant are also installed in this compartment—lubricating oil pumps, circulating water pumps, starting air flasks and starting air compressors. Here also are installed the transformers for the interior communication system, telephone and battery for the operation of the wireless installation and the nautical instruments.

REFRIGERATION INSTALLATION.

The refrigerating system is of the intercooling type, furnished by Linde-Atlas-Werke. It consists of two horizontal carbon dioxide twin com-

pressors, driven by twin steam engines. Each of the four compressors is of the two-stage type. The condensers are installed separately. They are steel tanks, internally protected against sea water. Individual groups of copper cooling coils may be shut off. Two motor driven centrifugal circulating water pumps, and three brine centrifugal pumps with two steam driven brine pumps as spares complete the installation. The arrangement of the ice machines, condensers and pumps is clearly shown on Plate I.

It was important for a good circulation to determine accurately the discharge pressure of the brine centrifugals, because the large number of cooling rooms is scattered not only up to sun deck or over 100 feet from the engine room floor but also throughout the length of the ship, thus requiring long pipe connections. With the exception of the ice-making room calcium chloride brine is used everywhere for cold transmission and two independent brine coolers are installed.

In addition to the electrical distant thermal control a control panel is installed in the ice-machine room which indicates by light signals the opening and closing of all doors of the cooling rooms.

FIRE FIGHTING APPARATUS.

The fire fighting installation on the *Bremen* is in keeping with the size of the ship and the high value of its machinery installation. In addition to a large number of hand fire extinguishers a special foam fire fighting installation was incorporated, consisting of a total of four separate foam generators. One foam generator is installed in the fireroom access, another in the ice-machine room and one in each of the main engine rooms. At each fireman's station there are two hose connections for the distribution of the foam, so arranged that a fire can be attacked from two sides.

The most important installation for fighting a large scale fire are the carbon dioxide fire fighting apparatus installed in each of the firerooms and the auxiliary engine room (Diesel), of such size that even the largest firerooms may be filled with carbon dioxide gas. The flasks containing the CO_2 gas are installed in a special room between the turbine room shafts on D deck connected to the firerooms by a common pipe. Each fireroom can be filled through a valve from the outside. The distributing system has outlets above and below the floor, the outlets having special non-clogging nozzles.

For water fire fighting there is a triple cylinder fire pump of 270 cubic meters per hour or 714,000 gallons per hour. The fire mains which cover the whole ship, may be also connected to the discharges of two deck washing, two sanitary and two bath water pumps, giving an additional amount at 1250 cubic meters per hour or 330,000 gallons per hour of water for fire fighting purposes.

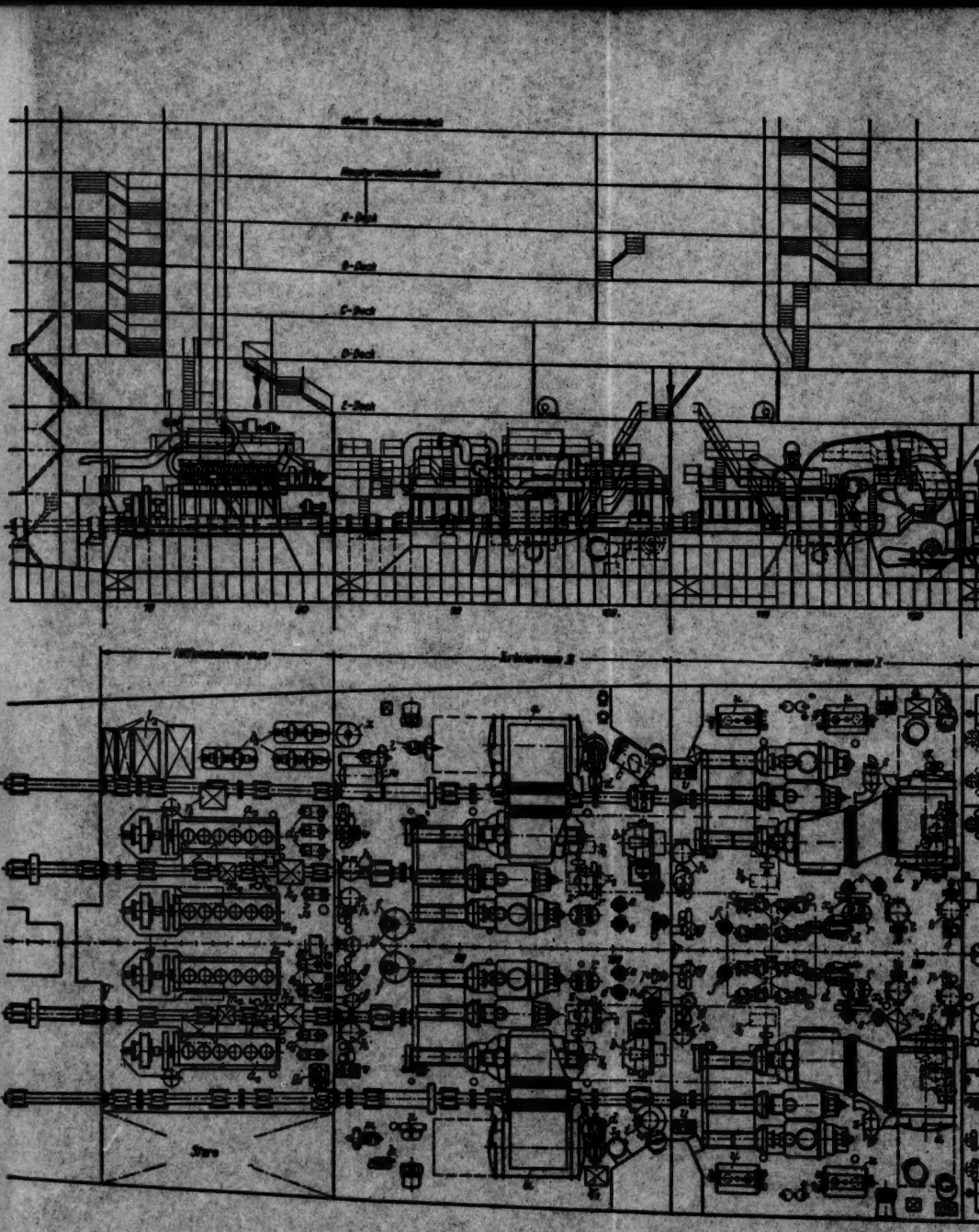
OTHER AUXILIARIES.

Other auxiliaries required by the ships extended facilities are as follows:

3 vertical simplex steam pumps of 75 cubic meters per hour (330 gallons per minute) for drinking water system.

4 vertical centrifugal pumps, turbine driven of 200 cubic meters per hour (880 gallons per minute) capacity each for supplying the required bath room water and for sanitary purposes.

2 vertical duplex pumps of about 200 cubic meters per hour (880 gallons per minute) capacity for deck washing.



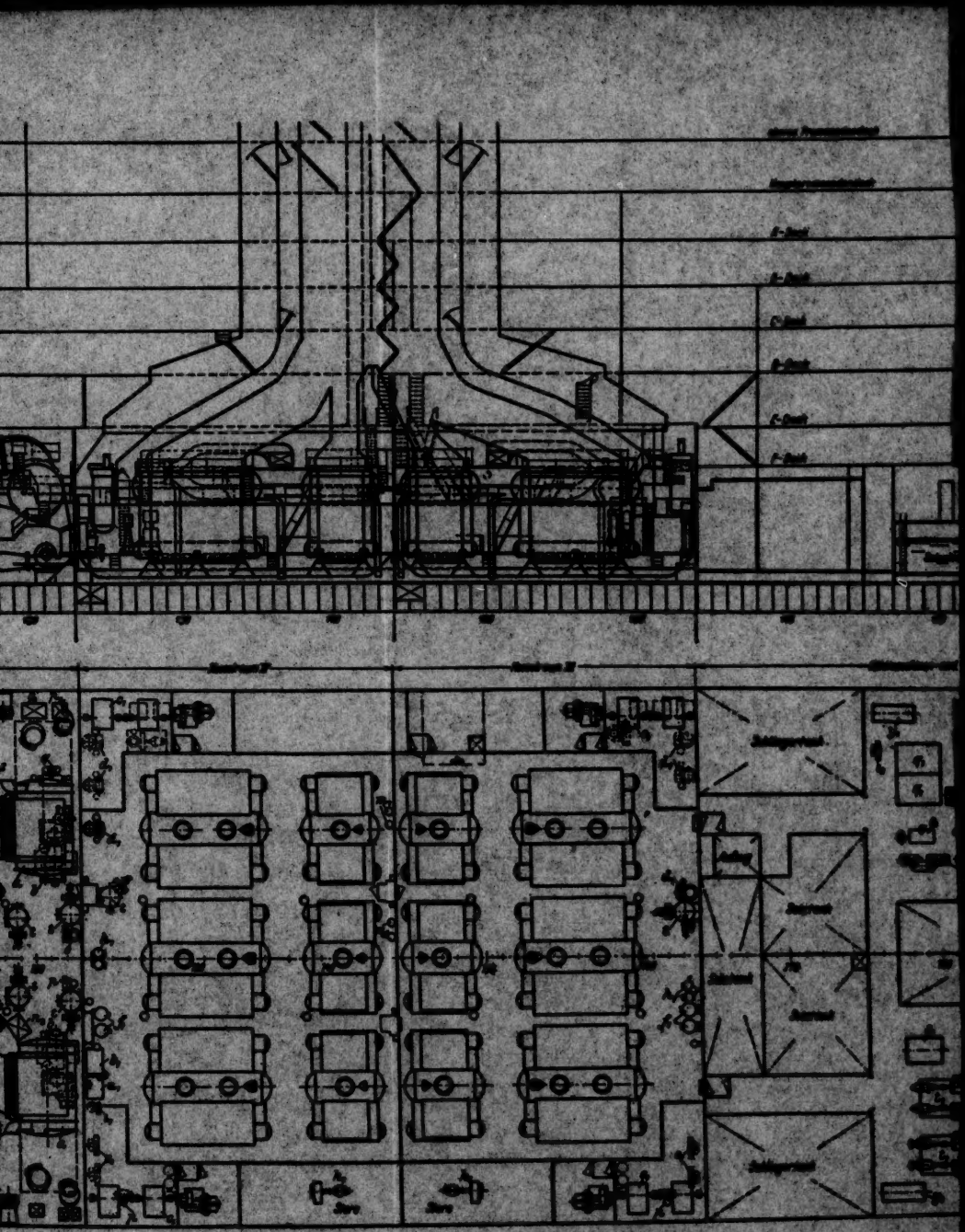
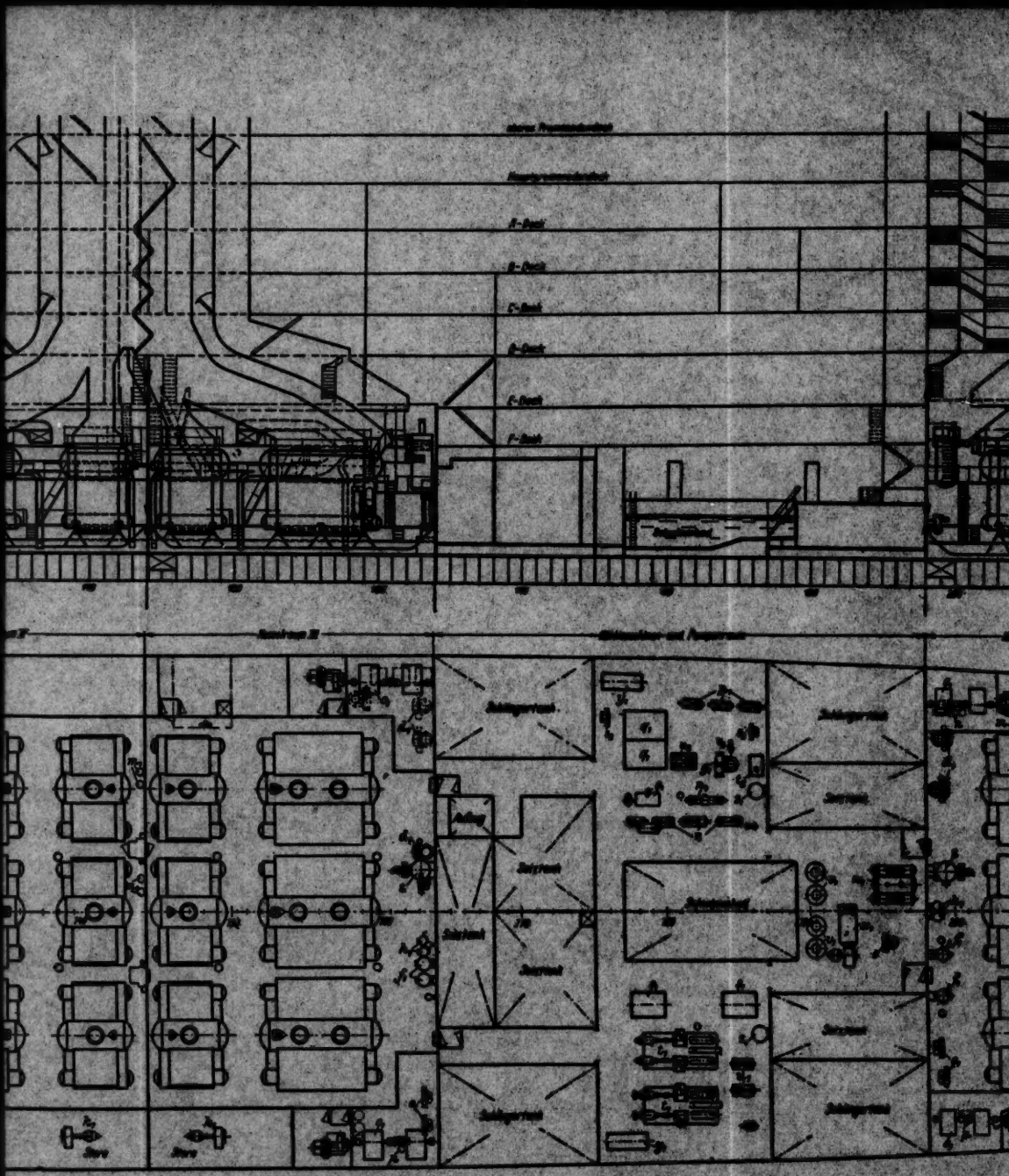


PLATE I.—ARRANGEMENT OF MACHINERY AND BOILERS.



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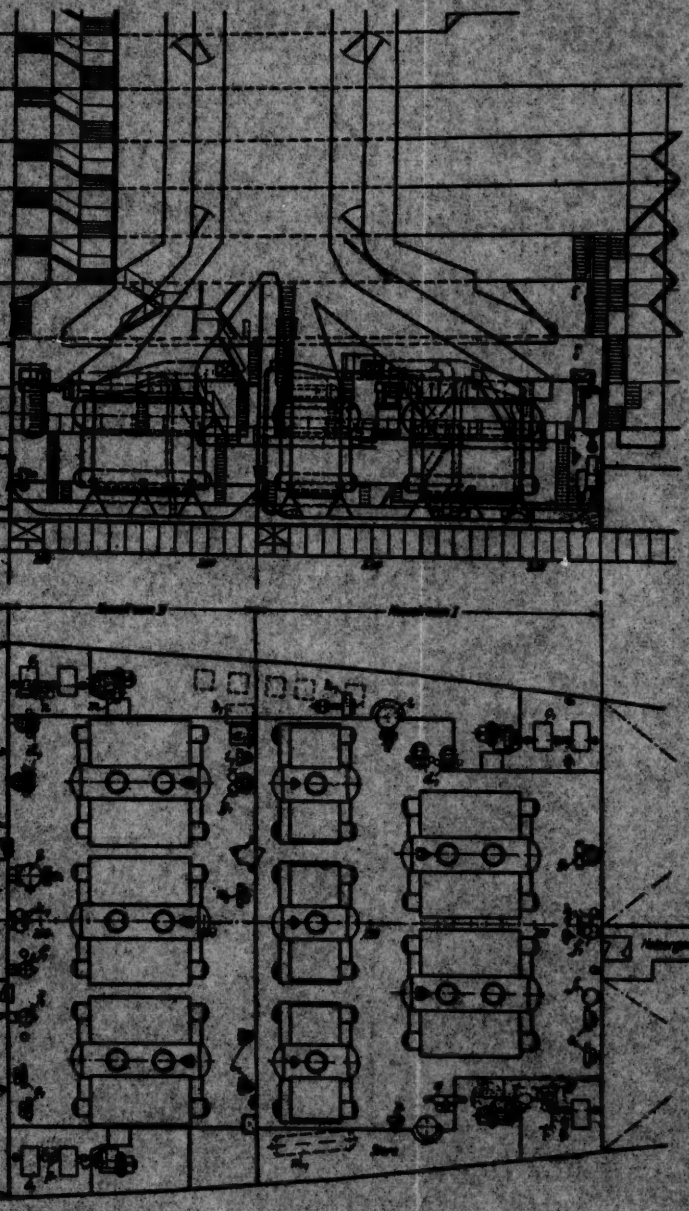


Hilfsmaschinenraum - Auxiliary
Turbinenraum - Main engine Room.
Kesselraum - Boiler room
Kühlmaschinen und Pumpenraum - Re
Schlingertank - Anti-rolling tank.
Setztank - Settling Tank.
Aufzug - Elevator.
Schwimmbad - Swimming Pool.
Durchgang- Fire Room access.

a - Main condensers.
b - Main circulating water pumps.
c - Auxiliary condensers.
d - Steam jet pumps.
e - Condensate and drain pumps
f - Main lub. oil pumps.
g - Lub oil filters.
h - Lub. oil coolers.
i - Bilge water oil separators.
k - Fire pumps.
l - Hand pumps.
m - Flushing pumps
n - Deck wash pumps.
o - Fresh water pumps.
p - Bilge pumps.
q - Ballast pumps
r - Turbine drain tanks.
s - Main feed pumps.
t - Feed water heater
u - Evaporators.
v - Lub. oil centrifuges.
w - Warm sea water installation.
x - Drain tanks for W.T. doors.
y - Pressure tanks for W.T. doors.
z - Main throttles.
a-1 - Float tank
b-1 - Separator tank.
c-1 - Fire room blowers.
d-1 - Aux. feed pumps.
e-1 - Fuel oil service pumps.
f-1 - Fuel oil transfer pumps.
g-1 - Fuel oil heaters.
h-1 - Pressure filters.
i-1 - Drain Pumps.
k-1 - Port fire room blowers.(for port
l-1 - Evaporator feed pumps.
m-1 - Air tanks in cool blower lines.
n-1 - Tanks
o-1 - Work Bench.
p-1 - Brine pumps.
q-1 - Brine coolers.
r-1 - Pumps for swimming pool.
s-1 - Condensers for ice machine.
t-1 - Ice machine.
u-1 - Circulating water pumps for ice m
v-1 - Fresh water filters.
w-1 - Warm fresh water installation.
x-1 - Aerators for swimming pool
y-1 - Tanks for dredge from cold storage
z-1 - Dirt collectors.
a-2 - Diesel generator sets.
b-2 - Transformers.
c-2 - Auxiliary air compressors.
d-2 - Circulating Water Pumps.
e-2 - Fuel oil service pumps.
f-2 - Auxiliary lub. oil pump.
g-2 - Lub. oil centrifuge.
h-2 - Ventilating fans.
i-2 - Starting air tanks.
k-2 - Lub. oil sump tanks.
l-2 - Lub. oil tanks.
m-2 - Drip oil tanks.
n-2 - Telephone/ berths.
o-2 - Lub. oil tanks.
p-2 - Cylinder oil tanks.
q-2 - Oil tank for bilge water oil sep
r-2 - Elevator for engine room personn
s-2 - Steam brine pumps.
t-2 - Salt solution and brine heater te
u-2 - Sludge collector for brine system
v-2 - Brine filter.
w-2 - Sludge pumps.
x-2 - Operating stands.
y-2 - Engine room ventilating fans.
z-2 - Cloth filter for bilge water oil

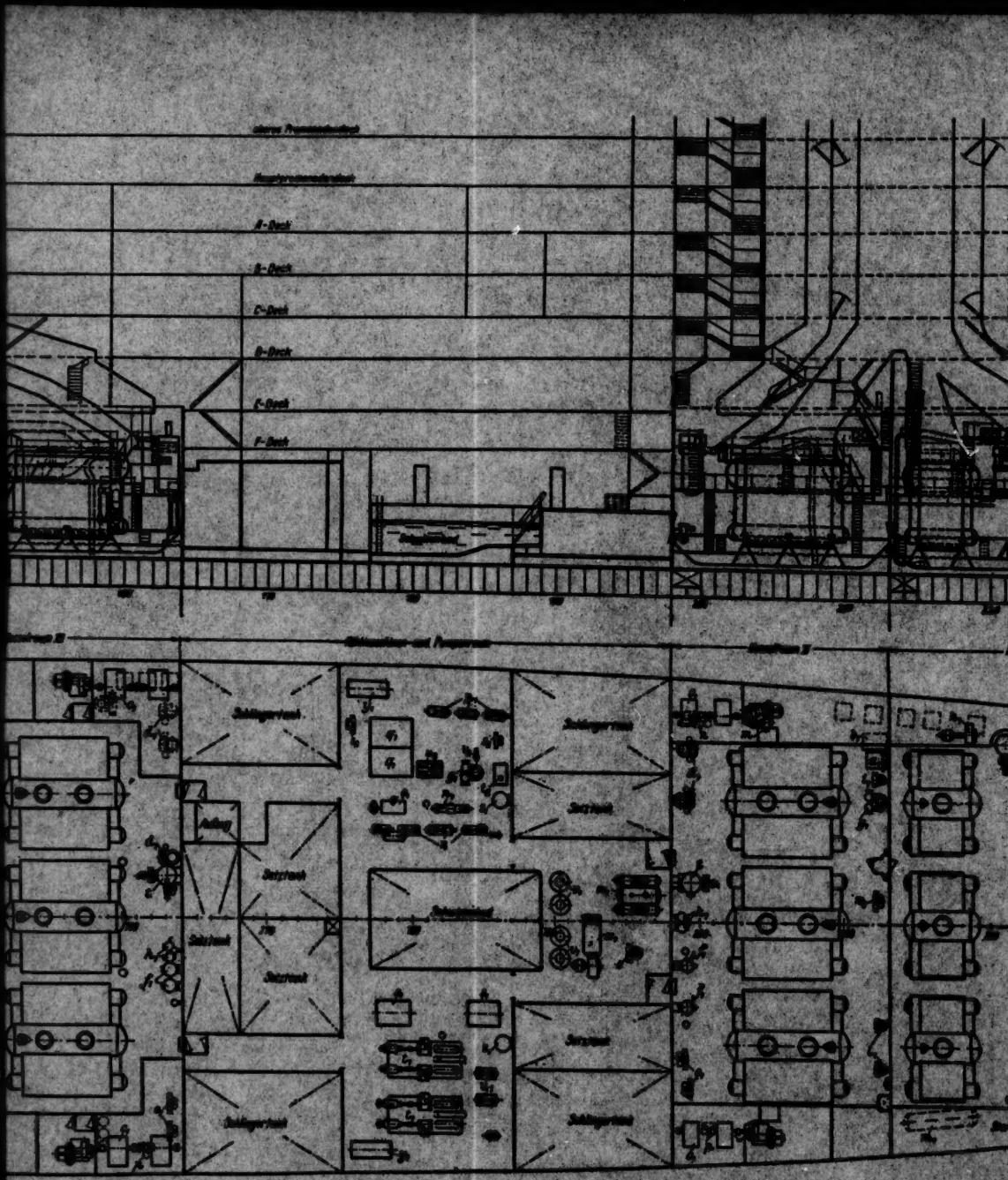
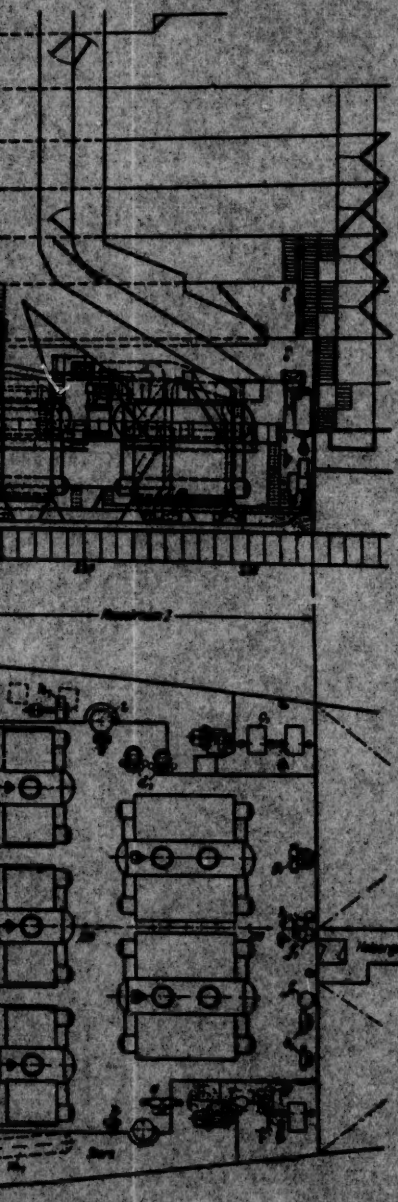


PLATE I.—ARRANGEMENT OF MACHINERY AND BOILERS.



Hilfsmaschinenraum - Auxiliary Engine Room
Turbinenraum - Main engine Room.
Kesselraum - Boiler room
Kühlmaschinen und Pumpenraum - Ice machine and Pump Room.
Schlingertank - Anti-rolling Tank.
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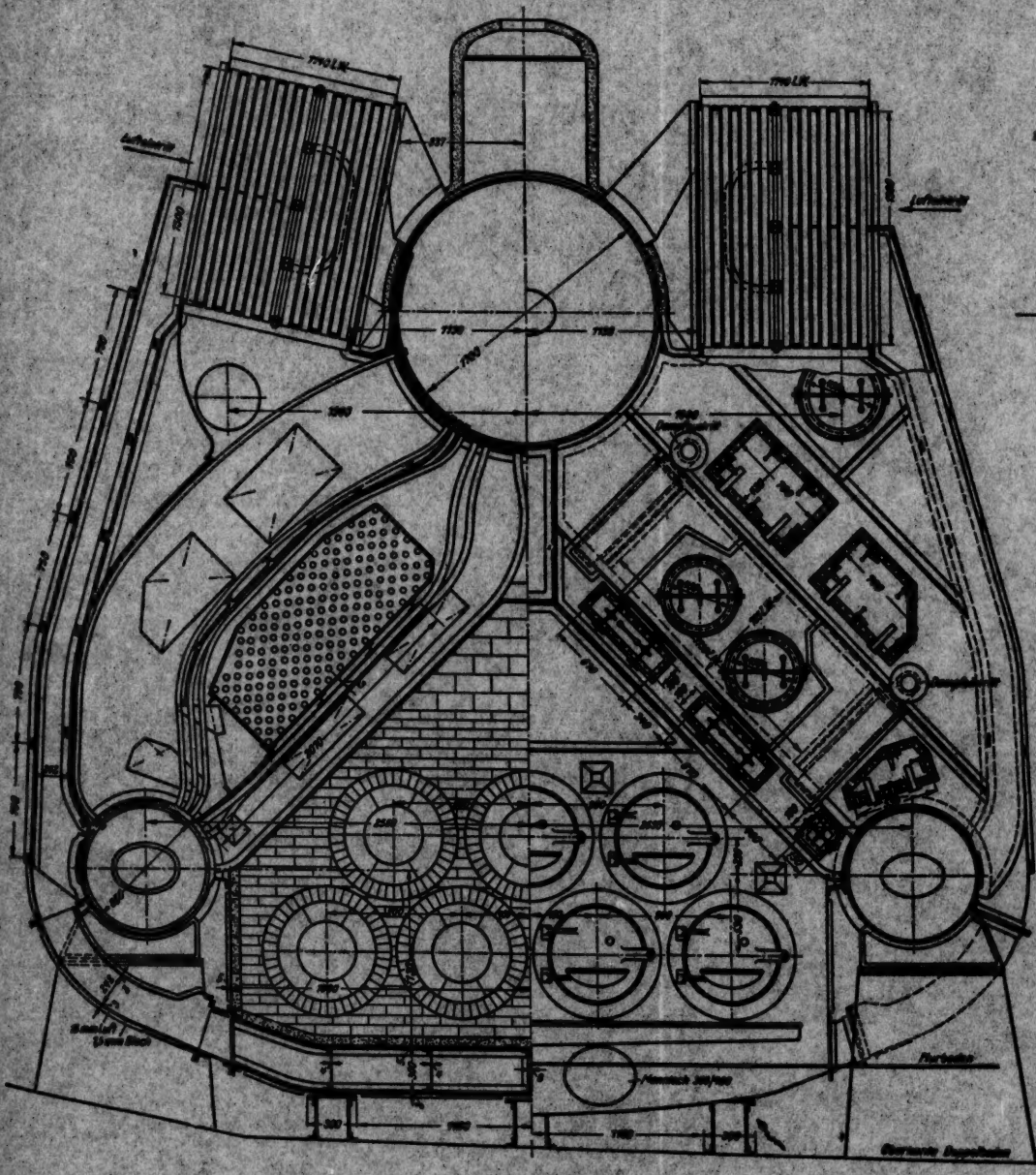
Schwimmbad - Swimming Pool.
Steigergang- Fire room access.

a - Main condensers.
b - Main circulating water pumps.
c - Auxiliary condensers.
d - Steam jet pumps.
e - Condensate and drain pumps
f - Main lub. oil pumps.
g - Lub oil filters.
h - Lub. oil coolers.
i - Bilge water oil removers.
k - Fire pumps.
l - Bath pumps.
m - Flushing pumps
n - Deck wash pumps.
o - Fresh water pumps.
p - Bilge pumps.
q - Ballast pumps
r - Turbine drain tanks.
s - Main feed pumps.
t - Feed water heater
u - Evaporators.
v - Lub. oil centrifuges
w - Warm sea water installation.
x - Drain tanks for W.T. doors.
y - Pressure tanks for W.T. doors.
z - Main throttles.
a-1 - Plant tank
b-1 - Separator tank.
c-1 - Fire room blowers.
d-1 - Aux. feed pumps.
e-1 - Fuel oil service pumps.
f-1 - Fuel oil transfer pumps.
g-1 - Fuel oil heaters.
h-1 - Pressure filters.
i-1 - Drain pumps.
k-1 - Port fire room blowers.(for port use)
l-1 - Evaporator feed pumps.
m-1 - Air tanks in soot blower lines.
n-1 - Lathe
o-1 - Work Bench.
p-1 - Brine pumps.
q-1 - Brine coolers.
r-1 - Pumps for swimming pool.
s-1 - Condensers for ice machine.
t-1 - Ice machines.
u-1 - Circulating water pumps for ice machines.
v-1 - Fresh water filters.
w-1 - Warm fresh water installation.
x-1 - Aerators for swimming pool
y-1 - Tanks for drains from cold storage rooms.
z-1 - Dirt collectors.
a-2 - Diesel generator sets.
b-2 - Transformers.
c-2 - Auxiliary air compressors.
d-2 - Circulating water Pumps.
e-2 - Fuel oil service pumps.
f-2 - Auxiliary lub. oil pump.
g-2 - Lub. oil centrifuge.
h-2 - Ventilating fans.
i-2 - Starting air tanks.
k-2 - Lub. oil sump tanks.
l-2 - Lub. oil tanks.
m-2 - Drip oil tanks.
n-2 - Telephone booths.
o-2 - Lub. oil tanks.
p-2 - Cylinder oil tanks.
q-2 - Oil tank for bilge water oil remover.
r-2 - Elevator for engine room personnel.
s-2 - Steam brine pumps.
t-2 - Salt solution and brine heater tank.
u-2 - Sludge collector for brine system.
v-2 - Brine filter.
w-2 - Sludge pump.
x-2 - Operating stands.
y-2 - Engine room ventilating fans.
z-2 - Cloth filter for bilge water oil remover.



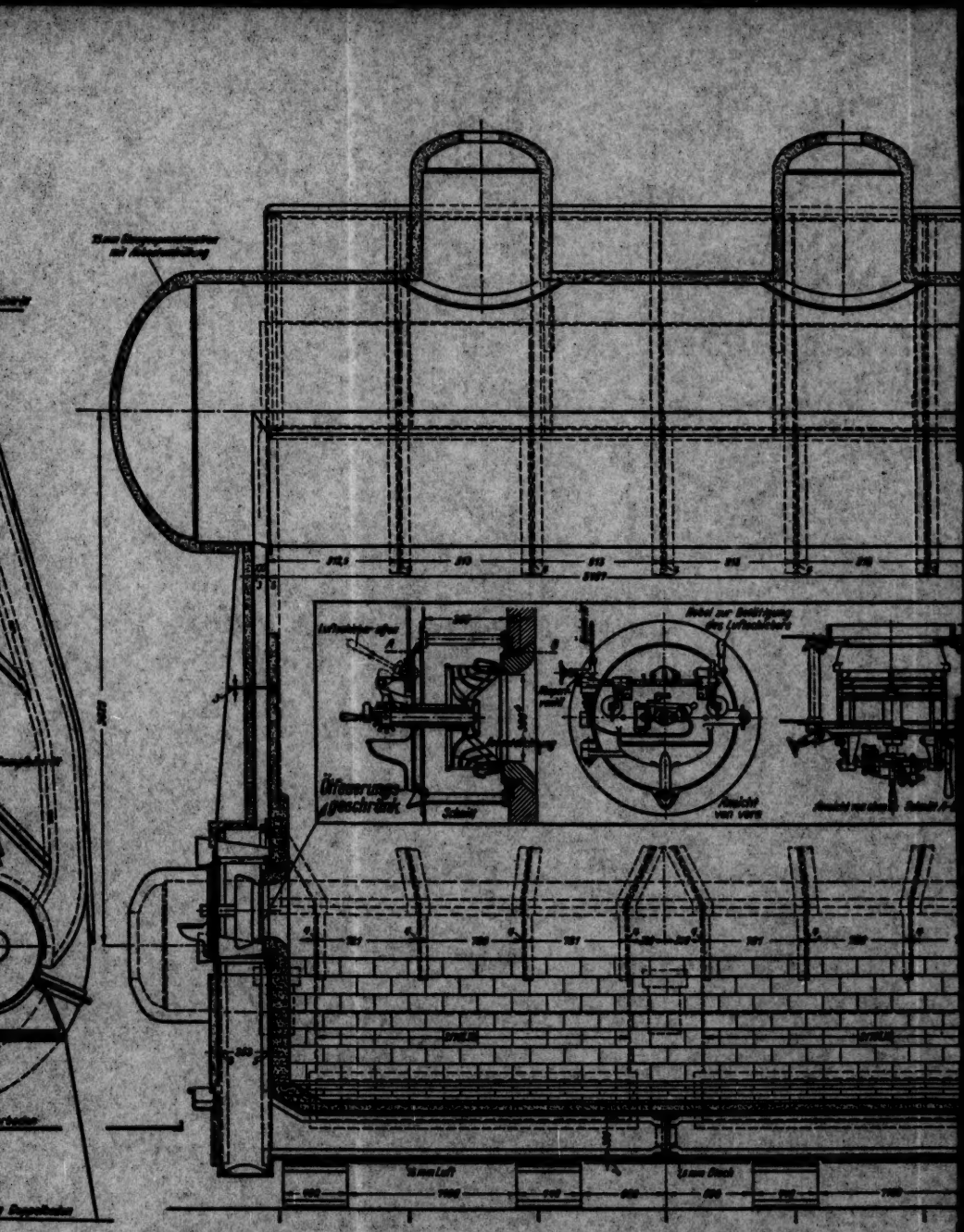


PLATE II.—WATER TUBE DOUBLE-END BOILER.

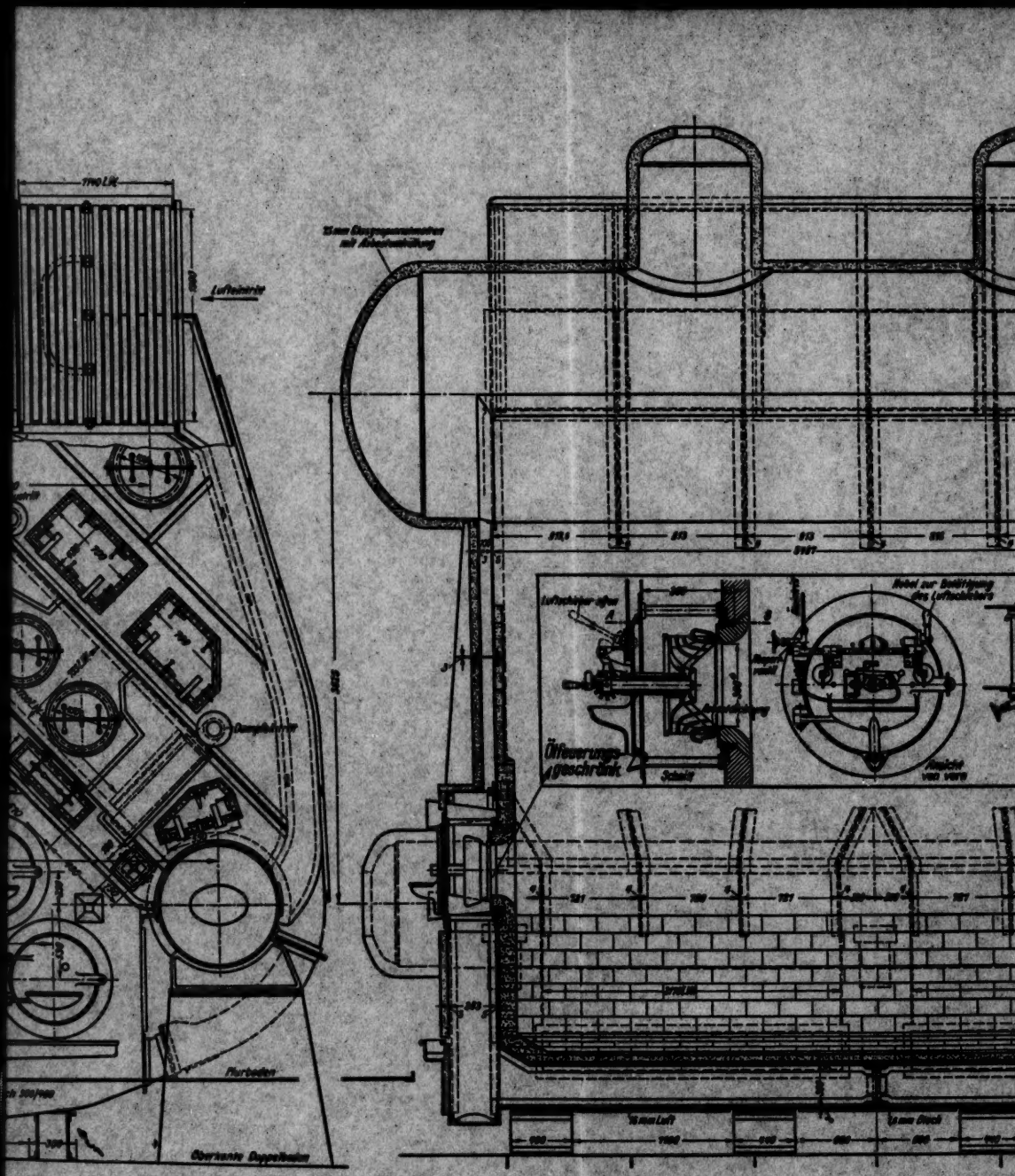
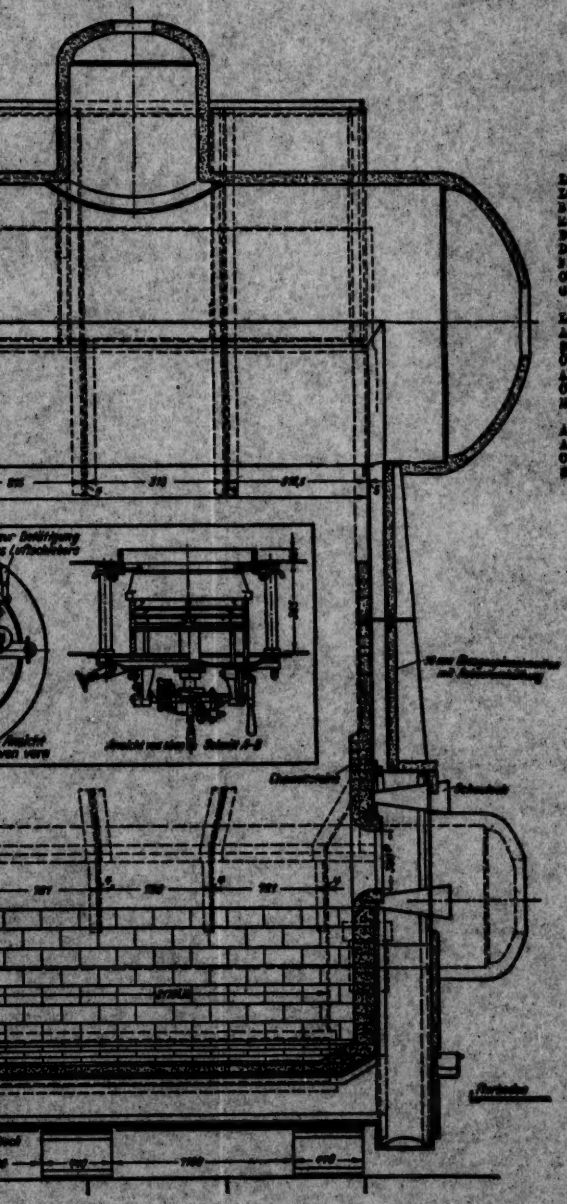


PLATE II.—WATER TUBE DOUBLE-END BOILER.



Luftzutritt - Air inlet.
Luft - air.
Blech - Sheet Steel
Mannloch - Manhole.
Dampfeintritt - Steam inlet.
Dampfaustritt - Steam outlet.
Flurboden - Floor.
Oberkante doppelboden - Upper edge of double bottom.
Glasabdeckung mit Asbestumhüllung - Spy glass with asbestos covering.
Luftschieber offen - Air valve open.
Asbestdichtung - Asbestos gasket.
Schnitt - Section.
Ölfeuerungs-gestänge - Oil burner linkwork.
Absperrventil - Shut-off valve.
Öleintritt - Oil inlet
Hebel zur Betätigung des Luftschiebers - Air valve operating lever.
Ansicht von vorn - Front elevation.
Ansicht von oben - Plan view.
Chamottesteine - Fire brick.
Mannloch - Peephole.

6 vertical simplex steam pumps of about 120 cubic meters per hour (530 gallons per minute) for bilge pumping.

2 vertical duplex steam pumps of about 250 cubic meters per hour (1,100 gallons per minute) as ballast pumps.

For port operation there is a special condenser installation provided, consisting of an auxiliary condenser of 165 square meters (1,775 square feet) cooling surface, one circulating water pump driven by a single-cylinder steam engine and a simplex wet air pump.

To supply compressed air for cleaning of boilers and operation of compressed air tools there are two vertical two-stage air compressors. Each of these air compressors is completely enclosed and has a common bedplate with a single cylinder enclosed type steam engine driving it.

To be able to pump overboard oily bilge water at any time it was desired, two bilge water oil separators of Turbulo type of a capacity of 150 tons per hour were installed—one forward and one aft.

For completeness sake a number of auxiliaries, pumps, etc., which are necessary for the swimming pool, must be mentioned:—One centrifugal pump for circulation and one for pumping overboard the water in the pool, also heaters for heating this water, aerators of water, etc., all of them independently operated.

Among the electrical auxiliaries may be mentioned the thermocouples to measure (*a*) live steam temperature at the throttles, (*b*) mixed steam temperature, (*c*) the superheated steam temperature at the superheaters, (*d*) combustion gas temperatures, (*e*) temperatures of auxiliary steam and (*f*) of exhaust steam.

Each thermocouple has its own measuring device. To obtain more accurate results it was not assumed as customary that the temperature and the cold junction temperature are about 20 degrees C. (68 degrees F.) but all the measuring instruments were calibrated for temperatures beginning with 0 degrees C. (32 degrees F.) and the cold junction temperature must be added to the temperature indicated.

The two steam whistles on the forward smoke pipe and the one on the after pipe are operated by electrical contacts from the bridge, each whistle having four such contacts. Suitably interlocking with the steam whistles on the forward smoke pipe is a motor driven fog horn. They are so interconnected that suitable fog signals, evenly spaced, may be made automatically either with the two steam whistles or with the fog horn. A special time contactor operates automatically the fog horn.

To inspect the CO_2 content of the combustion gases an electrical measuring device is installed in each fireroom. The salinity of water is controlled by a special device operating on the bridge method.

CONCLUSIONS.

The liner *Bremen* has already completed a number of round trips and proven the absolute reliability of its machinery and boiler installation. Besides this the builders may be proud of the very good economy achieved by this ship. Reliability and economy are, however, not only the results of efforts of the designers and builders, but are very much influenced by the attention received from the operating personnel. The thirty-six engineers of the *Bremen* deserve a great deal of credit for their share of the success.

FUEL AND LUBRICATING OIL AS THE CAUSE OF ENGINE TROUBLES.

By HERMANN JENTZSCH.

Translated by E. C. Magdeburger.

Abstract:—Discussion of operating difficulties of various internal combustion engines. Self-ignition and ignition factor. Explosive ignition as cause of knocking. Oxygen consumption for ignition and combustion. Thermal work of oxygen. Experiences with lubricating and fuel oils.

June 7, 1910, Roald Amundsen left Norway for a two months' experimental trip preliminary to his polar expedition. He was forced back, however, in three weeks by engine troubles. We read in Amundsen's book "Conquering South Pole," on page 192 of Volume I as follows: "Whether the fuel was not suitable or the engine was out of adjustment—the engineer did not know himself." In Volume II the captain of the *Fram* amplifies the above as follows: "Since the success of the whole south pole expedition may be dependent upon the faultless operation of the oil engine the continuation of the trip was given up and after three weeks the return trip to Bergen started. Here we exchanged the solar oil fuel for paraffin oil and from then on no difficulties in the operation of the oil engine have been experienced." The book also states that the "guilty" engineer was fired immediately upon return to Bergen. The new operator had assisted in the erection of the 180 H.P. Diesel engine of the *Fram* and suggested the use of a different fuel oil. The real "culprit" was now found—the 90,000 liters of the solar oil were now pumped out and a similar amount of paraffin oil was taken on board. With the new fuel the Diesel engine which was also "under suspicion" has completely vindicated itself during the long and arduous polar trip. Captain Kircheiss in his book "My Trip Around the World in the Fishing Vessel *Hamburg*" also relates a case of a hot-bulb motor, which otherwise performed faultlessly, going on a strike. The captain, however, was quite sure of the fitness of his operating engineer and well aware of the quality of his motor, so he quickly recognized in the fuel oil, which a smart Yankee salesman sold him dearly, the real "offender," and saved the situation with the purchase of a quantity of new oil.

Some time in 1923 a muffler explosion occurred during the starting of a Diesel engine on board a motorship, causing the loss of one human life. Investigation established no fault on the part of the engineer in charge, but pointed to self-ignition of small quantities of lubricating oil at about 300 degrees Centigrade, causing the explosion of unburnt gas mixture in the muffler. The author was commissioned to analyze the ignition properties of the oils used at the time. Both fuel and lubricating oils proved to be in accordance with specifications for their purchase, and it is only after constructing the ignition tester shown in Figure 1 (Julius Peters, Berlin N. W. 21 built the apparatus) that it was possible to prove that the accident was caused by the excessive *ignitability* of the lubricating oil. The method of operation of this device is as follows: Clean oxygen out of a container under pressure of one atmosphere passes through an accurately adjustable throttle valve and from it through a bubble counter and drier to a crucible made of a heat resisting and non-flaking steel and equipped with three ignition chambers and one thermometer chamber. The crucible is electrically heated and can be brought to any desired temperature. In determining

"the" *self-ignition point, i.e.,* the lowest temperature at which ignition will take place in a generous stream of oxygen, a drop of oil under investigation is placed into the small plate within the ignition chamber. The temperature t in degrees Centigrade, divided by the number of oxygen bubbles per minute b plus one, to take care of the oxygen reaching the ignition chamber from the outside air, was taken as the measure of ignitability or *ignition factor* Z. Thus

$$Z = \frac{t}{b+1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

To determine the ignition factor at any given temperature it is therefore necessary only to measure the quantity of oxygen required for ignition

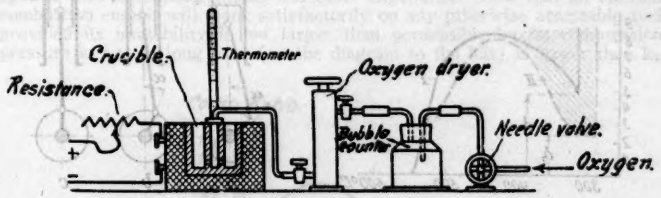


FIG. 1.—SCHEMATIC ARRANGEMENT OF IGNITION-FACTOR TESTER.

at that temperature. The ignition factor of the fuel in the above case was $Z_t = 6.8$ degrees, while for lubricating oil it was $Z_t = 9$ degrees, showing that the ignitability of the lubricating oil was 25 per cent in excess of that of the fuel oil. Investigation showed that the cylinders were very much out of round and ignition would not occur reliably while starting. Unburnt gas mixture was therefore expelled through the exhaust piping into the muffler and brought the small quantities of lubricating oil existing there to self-ignition temperatures. The outside temperature at the time of explosion was — 1 degree C. (30 degrees F.). There was no means of heating the engine room and the exhaust piping was filled with cold and therefore dense oxygen so that the passing unburnt gas mixture would absorb enough of it to satisfy the meager oxygen requirements of the lubricating oil present in the muffler for ignition. The ignition of the lubricating oil served to ignite the explosion of the gas mixture present and resulted in blowing away the muffler.

In 1924 a well-known German engine building firm had to furnish a number of aircraft engines to a foreign government. The engines performed satisfactorily on the test stand of the builder and were delivered to the customer for installation in the airplanes. Subsequent test flights showed that the engines either would not start at all or developed severe knocking when using the fuel furnished by the buyer and their acceptance was refused. The fuel investigations were successful only when the self-ignition curves of both fuels as shown on Figure 2, were determined by means of my ignition tester. It will be seen that the fuel II furnished by the buyer has a lower and larger pre-ignition area (shown hatched) than the fuel I used in the factory tests. The larger ignitability is also expressed by the higher ignition factor of this fuel which showed a ratio of 10 ÷ 5 as compared with the better fuel.

Empirical data shows that an oxygen stream of 120 bubbles per minute is approximately equivalent to oxygen concentration existing in the cylinder under 9 atmospheres = 128 pounds per square inch compression. Figure 2 shows that the fuel II would ignite in position k_1 of the piston, with a compression pressure slightly over 2 atmospheres = 28.4 pounds per square inch, while fuel I requires a compression of almost 5 atmospheres = 71 pounds per square inch. In starting, therefore, self-ignition would occur in position a and due to the large crank angle α the piston would be thrown backwards and could not perform any external work. When ignition for some reason (ignition lag, etc.) would first occur with a higher compres-

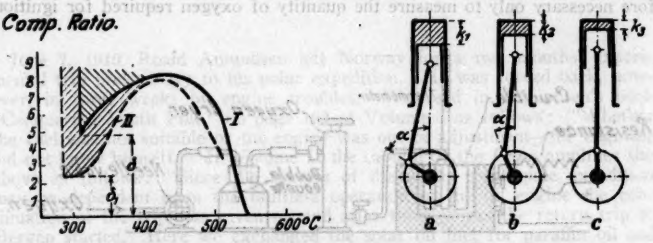


FIG. 2.—COMPRESSION PRESSURE AND KNOCKING FUEL DIAGRAMS.

sion k_2 shown in position b, corresponding to a smaller crank angle α, the crank could not any more be pushed backwards by the connecting rod. The action of ignition would be the same as in position c, *i.e.*, a well-known knock would be distinctly audible. The literature of the last few years is full of all kinds of explanations, frequently contrary to one another, of the causes of knocking. Let it be only mentioned here that the engine itself was also blamed for it. Tests made with an ignition pressure meter of my design* (DRP 404,810—H. Maihak A. G. Hamburg are builders of the apparatus) established the existence of *critical self-ignition temperatures*. It developed that throughout a range of temperatures beginning with self-ignition point and up to about 100 degrees C. above it under otherwise similar test conditions extremely strong ignitions would take place. In Figure 3 are shown igni-

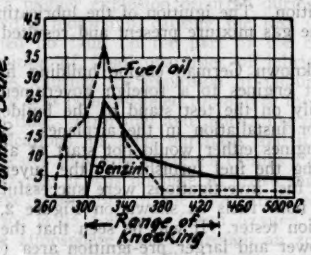


FIG. 3.—IGNITION-PRESSURE CURVES.

* See page 125 of booklet by the same author entitled: "Flüssige Brennstoffe V.D.I. Edition 1926.

tion pressure lines of a fuel oil and a benzine (gasoline). It would appear therefore that if such a noisy explosion-ignition takes place in positions b and c of Figure 2 the layer of lubricating oil on the wrist and crank pins will be squeezed out and the moving parts hit them with considerable force, creating a note comparable to the one produced by a light tap of a hammer on the anvil. When larger clearance existed around the pins the notes would be louder but not so clear. That the fuel is the sole cause of knocking is being gradually recognized everywhere, however, and numerous anti-knock substances, of which here only Benzol, Toluol, Ironcarbonyl and Tetraethyl lead shall be mentioned, have become known. Curve I in Figure 2 shows the improvement in the knocking gasoline as the result of addition of benzol. The above mentioned aircraft engines would now work satisfactorily even on fuel II, after sufficient benzol was added to it, so that there was nothing against their acceptance. This and other experiences show that an internal combustion engine will work satisfactorily on any otherwise acceptable fuel provided its ignitability is not larger than permissible by its compression pressure k_3 *i.e.*, as long as d (on the diagram to the left) is larger than k_3.

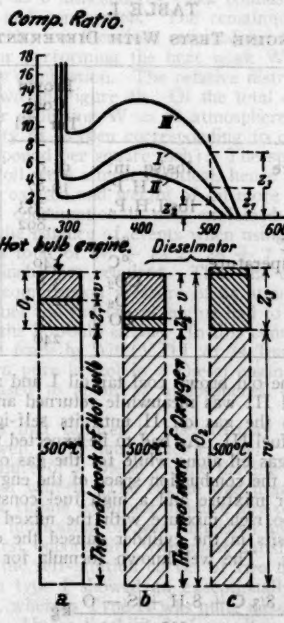


FIG. 4.—SCHEMATIC REPRESENTATION OF OXYGEN CONSUMPTION.

O—Total Oxygen Content.
w—Thermal Work of Oxygen.
Z—Ignition Work of Oxygen.
v—Combustion Work of Oxygen.

It may be mentioned here that in comparing ignition effect of the oxygen in the engine and in the test apparatus only actually measured compression pressures must be used. The compression ratio gives no accurate indication of the final compression pressure to be expected.

Interesting results were obtained from tests on a hot-bulb engine and on a Diesel engine of an older design. Both engines operated on a brown coal tar oil perfectly satisfactory. When once a new type of fuel was purchased for the hot-bulb engine, it stopped after 10 minutes of running on it and could not be started up again. Investigation established that the cylinder was completely covered up with soot and the valves would not close any more. The self-ignition curves for both fuels were then determined and drawn up (Figure 4) and they showed, that for the temperature existing in the hot bulb during the moment of injection (about 500 decrees C.), the new fuel oil represented by curve III, required twice as much oxygen for self-ignition as the old fuel which is represented by curve I ($Z_3 = 2Z_1$). After that experiments were made on the Diesel engine, the results of which are shown in Table I.

TABLE I.

DIESEL ENGINE TESTS WITH DIFFERENT FUELS.

		Brown Coal Tar Oil	Gas Oil	Gas Oil and Benzol
Mean indicated pressure	lbs/sq. in.	91.0	81.1	100.2
Indicated horsepower	I.H.P.	16.87	14.97	18.06
Fuel consumption	lbs/I.H.P.	.333	.448	.287
Specific gravity of fuel		.892	.870	.874
Ignition factor at 300°C	°C	8.3	16.6	4.6
Upper self-ignition temperature	°C	540	520	600
Exhaust gas analysis	CO_2	3.8	3.7	6.8
	O_2	12.2	14.0	6.2
	CO	.5	.65	5.4
Average R.P.M.		240	240	240

As fuel were used the old brown coal tar oil I and a mineral gas oil II. The unsatisfactory fuel III was meanwhile returned and so there was sufficient benzol added to the gas oil II until its self-ignition curve corresponded to that of the fuel III. As had to be expected there existed surplus of oxygen when using gas oil alone while for the gas oil mixed with benzol deficiency of oxygen in the combustion space of the engine was experienced. This resulted in a poor mixture and a high fuel consumption when using gas oil alone and a too rich mixture with the mixed fuel of gas oil and benzol when soot deposits in the cylinder caused the engine to stop after ten minutes of running. The well-known formula for air required

$$L = \frac{8/3\,C + 8\,H + S - O}{.232}\,kg \qquad . \quad . \quad . \quad (2)$$

shows that 1 kilogram of benzol required 1 kilogram less air than gas oil. The benzol-gas oil mixture, however, required about 4 atmospheres (56.8 pounds per square inch) more ignition pressure than the gas oil alone and 2.4 atmospheres (34.0 pounds per square inch) than the satisfactorily burnt brown coal tar oil. If even such a comparatively small difference in the

ignition work of the oxygen concentrated in the combustion space of engine cylinder could have had such important consequences, it would appear peculiar that the brown coal tar oil would operate satisfactorily both in the hot-bulb engine with a compression pressure of 6 atmospheres (85.2 pounds per square inch) and in the Diesel engine having a compression pressure of 34 atmospheres (483 pounds per square inch) Figure 4 shows schematically the explanation that I have arrived at which solves the problem.

In the hot-bulb engine the concentration of oxygen resulting from a compression pressure of 6 atmospheres (85.2 pounds per square inch) was evidently sufficient for satisfactory ignition and burning of brown coal tar oil. In Figure 4a this is represented by O. Since for the ignition only an ignition pressure of Z_1 equal to about 3 atmospheres (42.6 pounds per square inch) was required, the rest of the oxygen corresponding to a pressure $V_1 = 6 - 3 = 3$ atmospheres (42.6 pounds per square inch) was left for combustion. When using the same fuel in the Diesel engine Z_1 and V_1 had to remain the same and from the total concentration of oxygen corresponding to compression pressure $O_2 = 32$ atmospheres (483 pounds per square inch) only $Z_1 + V_1 = 6$ atmospheres (85.2 pounds per square inch) were needed for ignition and combustion. The remaining 28 atmospheres (398 pounds per square inch) are therefore required for the same purpose as the hot-bulb, namely, for performing the heat work W and are not necessary for either ignition or combustion. The relative distribution of oxygen when using gas oil is shown on Figure 4b. Of the total concentration of oxygen O_2 there is left after deducting $W = 28$ atmospheres and $Z_2 =$ about 1 atmosphere the quantity of oxygen corresponding to compression of 5 atmospheres (71 degrees pound per square inch). The specific weights of gas oil and brown coal tar oil were almost identical, hence to obtain an equal ratio between combustion oxygen and weight of oil more gas oil had to be added and the higher fuel consumption is thus explained.

Figure 3c shows the balance of events when using the fuel III. For heat work and ignition a total of $W + Z_3 = 28 + 7 = 35$ atmospheres (497 pounds per square inch) are required. Thus the concentration of oxygen corresponding to a compression pressure of 34 atmospheres is not sufficient for ignition of the fuel and the Diesel engine had to stop for lack of oxygen in the same way as the hot-bulb engine. In this connection I wish to remind of the announcement made by Alt (V.D.I. of 14 June, 1923) that the Krupp Company could burn pure benzol in a Diesel engine when the compression pressure was increased to about 45 atmospheres (640 pounds per square inch).

Experience shows that failures to operate may be caused by unsuitable lubricating oil as well as by unsatisfactory fuel. As an instance may be cited the sudden failure of a Diesel engine driving a generator on board a motorship in tropical waters. The engine was first blamed for the failure, but inspection of the engine could not produce any plausible cause of failure. Finally a change of lubricating oil was suggested and a new type of oil was purchased at the first opportunity with most satisfactory results.

Figure 5 shows the self-ignition curves of two different brands of lubricating oil, of which type I showed an exceptionally high resistance to the influence of oxygen, whereas Type II was quite unsuitable for use in internal combustion engines. Under "stability" of an oil is understood its ability to stand up under the influence of time, oxygen and heat. Experience shows that an otherwise satisfactory oil is the more stable the higher its self-ignition curve and the lower its ignition factor. Unstable lubricating oils could under certain circumstances, as for instance in air-cooled engines for aircraft, noticeably interfere with their operation and even completely stop them. When an aircraft motor is being forced, either by flying against the

wind, failure of one of the motors in a multi-motored plane, etc., the required temperature of about 250 degrees may be easily reached while the necessary small quantity of oxygen can be taken out of the passing combustion air. On test of a Diesel engine the ignition factor of an unstable oil rose from 8.5 degrees to 12.5 degrees at 300 degrees C. after three hours of operation.

Investigation of an explosion in a discharge piping of an air compressor installation and another explosion in the starting air piping of a marine Diesel engine three days afterwards established beyond a doubt that lubricating oil was the cause of both failures.* The same lubricating oil was used in both cases and from its high ignition factor—9.65 degrees, a very unstable character of the oil had to be expected.

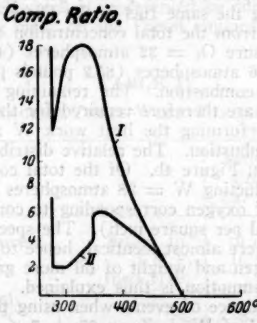


FIG. 5.—SELF-IGNITION CURVES OF LUBRICATING OILS.

At the annual meeting of the Brennkrafttechnischen Gesellschaft (Combustion Engineering Society) Mr. Auffhäuser declared that there is essentially no difference between the combustion phenomena in the Diesel engine and in a furnace. From my own experiments through many years I came to the conclusion that the ignition phenomena of combustible substances require the same conditions of environment. In the above a number of proofs for the justification of this view have been cited, relating to Diesel hot-bulb and explosion type engines. Other experiments which will be reported later† showed the existence of similar requirements for using coal dust in Diesel engines as in boiler furnaces. About the accidents with oil fired boilers, which came to my notice, the following can be stated.

A steamer with oil fired boilers received instead of an easily ignitable oil an oil with a high ignition oxygen consumption. As the result immediately after lighting off the boiler an explosion occurred in the upper reaches of the boiler interior which destroyed the uptakes. The cause of the accident must be seen in the fact that the existing blower installation was not sufficient to furnish the oxygen required for ignition of a difficult ignitable fuel oil, resulting in oil fumes gathering up in pockets in the upper portions of the boiler interior and finally exploding.

In another oil fired boiler installation a change of fuel oil was also made with resultant inability of the boiler to furnish enough steam for the engine

* See page 147 of the above-mentioned booklet.

† Published in "Werft-Reederei-Hafen of 22 Oct., 1929, on p. 409.

it was successfully operating prior to that. Investigation by means of my ignition tester showed that the self-ignition temperature of the new fuel was 430 degrees C. and the lowest ignition factor was 2.2 degrees whereas the fuel oil formerly used had 260 degrees C. self-ignition temperature and an ignition factor of 13 degrees. The cause of trouble was therefore to be found in the fact that the existing blower installation which was large enough to furnish the oxygen required for ignition and combustion of the fuel originally used was not sufficient to do the same for the new fuel oil. The very high self-ignition temperature means also that this 170 degrees difference was formerly used for steam generation instead. After purchasing fuel with a high ignition factor and low self-ignition temperature the former maximum power of the engine could again be developed. The fuel oil best suited for the use under any boiler is therefore such an oil which has in addition to high calorific value, *i.e.*, high content of heat producers, also a high ignition factor or economy in blower work through better utilization of oxygen in the air furnished by the latter.

Conclusions. It may be assumed as proven that the finding of the cause of the trouble in the cases above cited would have been difficult without the knowledge of self-ignition characteristics of the fuels used. On the other

TABLE II.

SELF IGNITION TEMPERATURES AND IGNITION FACTORS.

No.	Substance	Where Used	Self-Ignition Temperature °C	Ignition Factor °C
1	Gasoline	Explosion motors with 28–42 pounds per square inch compression	270–300°	At 300°C up to 9°
2	Gasoline (doped)	Explosion motors up to 64 pounds per square inch compression	270–315°	At 300° up to 5°
3	Gasoline (doped)	Explosion motors up to 100 pounds per square inch compression	300–325°	At 300° up to 3.5°
4	Benzol	Explosion motors up to 156 pounds per square inch compression	560–600°	At 600° up to 6°
5	Fuel Oil	Airless injection Diesel	260–300°	At 500° 50 to 80°
6	Fuel Oil	Air injection Diesel	260–300°	At 500° 14 to 18°
7	Fuel Oil	Hot bulb engine	260–300°	At 500° 14 to 18°
8	Lub. Oil	Internal Combustion Engine, and L. P. Air Compressors	260–300°	At 300° up to 5°
9	Lub. Oil	H. P. Air Compressors and H. P. Cylinders	260–300°	At 300° up to 2°
10	Transformer Oil	Oil Switches and Transformers	250–280°	At 300° up to 6°
11	Boiler Oil	Marine boilers	260–290	The higher the better

hand human lives would have been saved and much damage to machinery prevented if suitable apparatus to determine the requirements in heat and ignition oxygen of the fuel to be used were utilized. The above quoted old formula for air required is being used everywhere even now, despite the fact that every automobile and motorcycle operator knows well enough from everyday experience that the much smaller amount of air required by benzol as given by this formula, compared with gasoline, is contradictory to actual facts—at the same time my theory explaining the facts by the difference in the required ignition oxygen, is making but very slow progress. It may be mentioned that the human resistance against anything new, which upsets the prevailing opinions and theories, explains no doubt this phase of the matter. Since, however, the above-cited practical examples show that it is possible by investigating extremely small quantities of oil and thereby establish the causes of many accidents, a few numerical data on different oils used as fuels and for other purposes are given in Table II.

In using the above figures it must be borne in mind that experience shows that aluminum alloy pistons in explosion motors have a retarding catalytic effect upon the self-igniting properties of a fuel. When using light metal pistons it is therefore customary to credit the fuel with from 15 to 30 pounds per square inch more compression than their ignition factor would indicate. A tolerance of ± 1 degree may be always allowed. Finally it must be added that knowledge of self-ignition properties of oil furnishes no clew whatever to any of its physical and chemical characteristics and determination of self-ignition data does not dispense with the determination of the latter. If it is necessary, however, to find whether engine, fuel or lubricating oil was the cause of an operating trouble, then in most cases all other methols of investigation are usually inadequate.—"Werft-Reederei-Hafen," June 7, 1929.

THE LARGEST BRITISH MOTORSHIP.

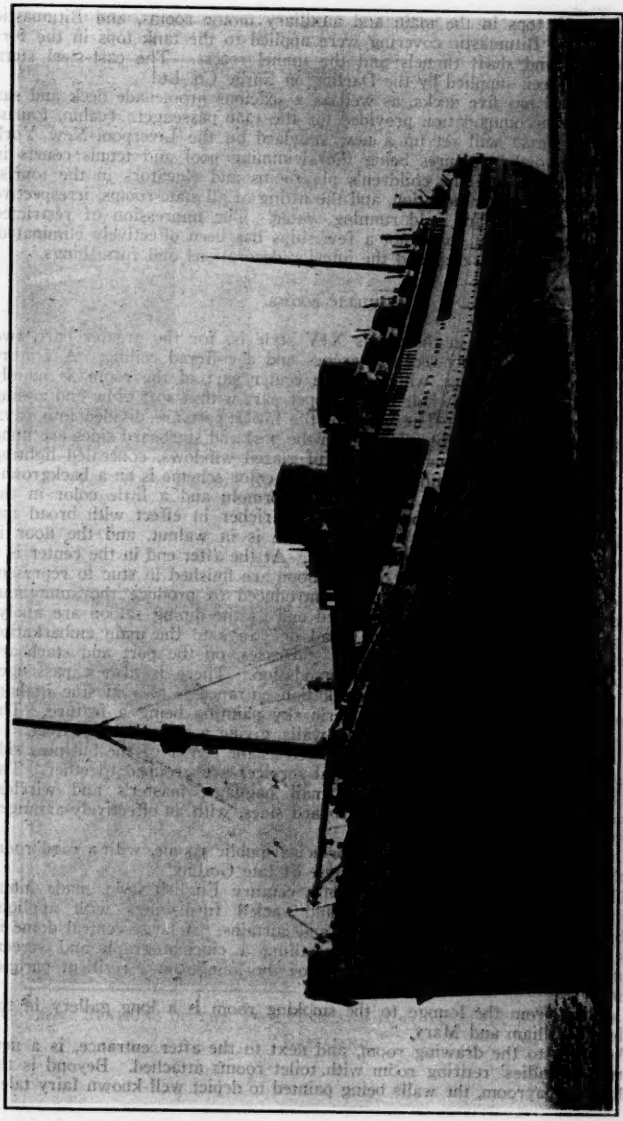
COMPLETION OF THE 27,000-TON WHITE STAR LINER "BRITANNIC," FOR THE LIVERPOOL-NEW YORK PASSENGER SERVICE. BUILT AND ENGINED BY HARLAND & WOLFF LTD., BELFAST.

The White Star liner *Britannic* is the most notable motorship to be completed this year. A 27,000-ton twin-screw vessel, she is the largest motorship ever built in a British yard, and is also the largest British-owned Diesel-engined vessel. Furthermore, she is the second largest motorship in the world, being exceeded in both tonnage and power by the quadruple-screw N.G.I. passenger motorship *Augustus* only.

The *Britannic* has been built and engined by Harland & Wolff Limited, Belfast, for the Liverpool-New York service of the White Star Line. She has been built under the survey of Lloyd's Register and the Board of Trade and is provided with a straight stem, cruiser stern, and two low, broad funnels. Her principal particulars are as follow:—

Length, between perpendiculars, feet	680
Breadth, moulded, feet	82
Depth, moulded, feet and inches	43-9
Gross tonnage, tons	26,943
Service speed, knots	16/17

The ship is subdivided into 13 watertight compartments, and there is a continuous double bottom arranged to carry fresh water, water ballast and oil fuel. Bitumastic enamel and Bitumastic heat-proof cement were applied



NEW WHITE STAR LINE'S MOTORSHIP "BRITANNIC" AT ANCHOR IN BELFAST LOUGH.

to the tank tops in the main and auxiliary motor rooms, and Bitumastic solution and Bitumastic covering were applied to the tank tops in the forward pipe and shaft tunnels and the tunnel recess. The cast-steel stern frame has been supplied by the Darlington Forge Co. Ltd.

The vessel has five decks, as well as a spacious promenade deck and sun deck. The accommodation provided for the 1550 passengers (cabin, tourist and third class) will set up a new standard on the Liverpool-New York route, noteworthy features being the swimming pool and tennis courts in the cabin class, and the children's playrooms and elevators in the tourist and third-class accommodation, and the fitting of all state rooms, irrespective of class, with hot and cold running water. The impression of restricted space so common on board not a few ships has been effectively eliminated. Heaton Tabb & Co. designed the interior decorations and furnishings.

PUBLIC ROOMS.

The dining saloon in the Louis XIV style is, for the greater part, two decks high, with deeply-recessed arches and a coffered ceiling. A feature is made of the lighting, which, in the center part of the room, is mainly concealed, flooding the whole of the upper part with a soft glow and adding to the apparent height. The walls of the lower part are divided into bays, with broad pilasters. The windows on the port and starboard sides are hung with lambrequins framing the graceful glazed windows, concealed lighting behind these giving a sunlight effect. The color scheme is on a background of old ivory colored stuc, relieved with ormolu and a little color in the panels, the central dome being somewhat richer in effect with broad flat wall spaces as a contrast. The furniture is in walnut, and the floor is arranged in a quiet and dignified design. At the after end in the center is a large cold buffet. The walls of the saloon are finished in stuc to represent masonry, with just sufficient color introduced to produce the sumptuous effects of the period. At the forward end of the dining saloon are finely-proportioned glazed screens and broad doorways to the main embarkation entrance from which rise two wide staircases, on the port and starboard sides, to the main passenger accommodation. There is also a passenger lift to the other decks. The embarkation entrance is also in stuc in harmony with the saloon, an atmospheric sky painting being a feature. The whole is lit with amber bowls on the walls, giving a soft glowing effect.

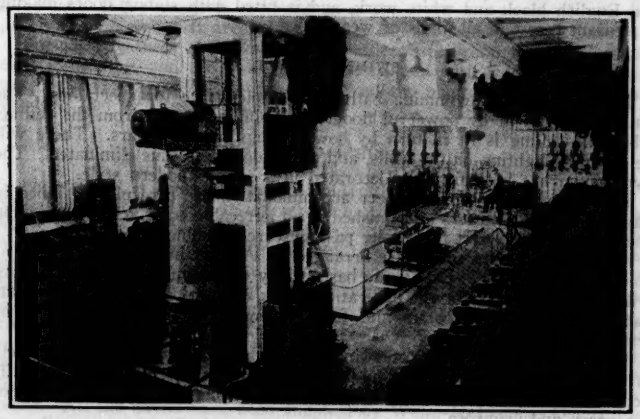
The B deck entrance has been arranged as the center of the business side of passenger life, and all the essential services are grouped together. The purser's, chief steward's, enquiry, mail baggage, master's and wireless offices are here on the port and starboard sides, with an effectively-arranged shop on modern lines near the lift.

On the promenade deck are the principal public rooms, with a card room at the forward end in the French style of late Gothic.

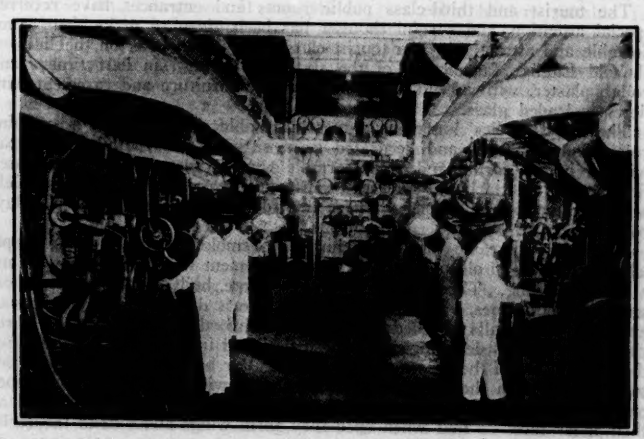
The lounge is in the late eighteenth century English style, made interesting with modern pier glasses and varied furnishings with applique embroidery in the rose pink pelmets and curtains. A large central dome in this room gives opportunity for installing a cinematograph and screen. Dancing will take place in the center of the lounge on a resilient parquet floor.

Leading from the lounge to the smoking room is a long gallery in the style of William and Mary.

Adjacent to the drawing room, and next to the after entrance, is a new feature—a ladies' retiring room with toilet rooms attached. Beyond is the children's playroom, the walls being painted to depict well-known fairy tales.



TOP MAIN ENGINE ROOM PLATFORM OF THE "BRITANNIC."



CONTROL STATIONS OF THE TWO HARLAND-B. & W. MAIN ENGINES OF THE "BRITANNIC."

The smoking room is a reproduction of a Tudor room in an Elizabethan mansion. The veranda café at the after end of the promenade deck is in old English black and white work, and is fitted with sliding doors making a pleasant resort in all kinds of weather.

SWIMMING BATH.

There is a large swimming bath with ample space for spectators. The bathing pool is lit by powerful flood lights shining direct from the ceiling, while the ceiling is lit in turn by bowl fittings on the pillars. There will be no direct lighting on the walls, and all shadows are thus eliminated. The gymnasium has been fittingly decorated in a simple modern style.

The alleyways to the passenger accommodation are carefully designed in bays divided by pilasters and finished in an agreeable color scheme, avoiding the uninteresting effect these long passages are so liable to have.

A number of suite rooms are arranged on A deck in varying schemes of color and decoration. Each suite has a lobby connecting the various rooms, with a useful recess for the innovation trunk now so popular, and a private bathroom panelled in colors and fitted up with all that is demanded by travellers.

A number of other suites and single cabins are also provided in a variety of styles and color schemes. The private bathrooms on A deck are specially decorated and fitted out, the suite bathrooms being treated in schemes of sea green and primrose.

All the state rooms, cabin, tourist and third class, have hot and cold fresh water, and the cabin baths have hot and cold showers.

TOURIST AND THIRD CLASS.

The tourist and third-class public rooms and entrances have received special attention, and careful thought has been given to make them comfortable and attractive. The tourist-class smoking room is on the lines of the old English farmhouses of the sixteenth century, in half timber and rough plaster, with an old tiled floor, antique furniture and casement windows in leaded glass.

The tourist ladies' lounge is a most agreeable room, painted a sunshine color, and furnished and decorated as a sitting room, with a large bay window in one corner and a comfortable window seat. A deeply-recessed fireplace on the starboard side, and comfortable easy chairs and chesterfield settees, complete the scheme. There is also a tourist-class children's playroom.

The interchangeable tourist or third-class smoking room is in a simple scheme of oak and old plaster with quaint casement windows, old settles and a beamed ceiling. The doors are of the old type, built of planks, with bold wrought-iron hinges. The tourist-class lounge is designed in a more modern style, with walls panelled in polished hardwood. The tourist- or third-class lounge on B deck forward is also designed in the modern style. For the convenience of the passengers a Catholic altar has been arranged between two bookcases in the center of the after end, with folding doors to open out revealing a complete altar recess, the folding doors being decorated on their inner faces so that when open they form wings and extend the ecclesiastical designs on each side.

The tourist- or third-class lounge on C deck aft, while not so large as the lounge on B deck forward, nor designed in so formal a manner, is a pleasantly-painted room with gay curtains to the port lights, amber-colored electric light fittings on the ceiling, and Ruboleum floor.

The third-class smoking room is designed in old oak and plaster, and has an old-world appearance with rough hewn oak timbers on the walls. The third-class ladies' room is designed as a simple drawing room, with wall panelling painted a soft green color, and a dado in a deeper shade.

The chief features of the third-class children's playroom are the concealed electric lighting, and the walls which are painted with subjects from many nursery rhymes. A Wendy hut carries out the playroom idea, with quaint windows and picturesque roof.

DECK MACHINERY.

The vessel has four large cargo holds; the various derricks, supplied by Stewarts & Lloyds Limited, are served by twelve 3-ton and four 5-ton Laurence, Scott electric winches. These are of the latest worm-geared type, with contactor controllers, to be turned out from the Norwich works of Laurence, Scott & Electromotors Limited, who also supplied the 20 electric boat-lowering winches used in connection with the Welin-McLachlan quadrant type davits. The particulars of the various lifeboats and davits are as follow:—2 sets handling 25-foot emergency lifeboats; 8 sets each handling two 30-foot superimposed 86-person lifeboats; 4 sets handling 30-foot single 86-person lifeboats; 2 sets handling 28-foot 30-person motor lifeboats; 3 sets handling 26-foot 42-person lifeboats; 2 sets handling 27-foot 45-person lifeboats. All the davits have been supplied by Welin-McLachlan Davits Limited, Glasgow. W. Gradwell & Co. Ltd, Barrow-in-Furness, have supplied 25 sections of their foreign-going buoyant apparatus, 6 feet 7 inches by 4 feet 7 inches, each being certified to carry 22 persons.

Like the cargo and boat-lowering winches, the windlass and capstan gear is operated electrically, and is of the vertical type, manufactured by Napier Brothers Ltd., Glasgow. It consists of two cable holders suitable for working the 330 fathoms of 3 7/16 inch stud link chain cables, which was supplied by N. Hingley & Sons Ltd., and two warping capstans, which are driven from the deck below by two 170-B.H.P. Laurence Scott electric motors through spur, worm and bevel gearing. The full load pull at either cable holder is 60 tons at 20 feet per minute, and at either capstan 20 tons at 60 feet per minute, while the slack rope speed is 250 feet per minute. Normally, the port set is independent of the starboard set, but the gear is so arranged that, if desired, the port motor can drive the starboard cable holder and warping capstan, or *vice versa*. Each cable holder is fitted with a Napier automatic differential self-holding brake. The holders, capstans, soleplates and gear wheels are made of cast steel, the gear wheels having machine-cut teeth. The electrical equipment in connection with the windlass and capstan gear has been supplied by Laurence, Scott & Electro-motors Limited, the control being on the Ward-Leonard system. The three bower anchors, each weighing 10 tons, and 330 fathoms of stud link cable 3 7/16 inches, weighing about 100 tons, were supplied by N. Hingley & Sons Ltd., while the stockless type forged steel anchors are by Samuel Taylor & Sons (Brierley Hill) Ltd.

The two electric after warping capstans have also been supplied by Napier Brothers Ltd., each capstan being driven by a 100-B.H.P. electric motor through worm and bevel gearing. The full load pull of either capstan is 20 tons at 50 feet per minute, while the slack rope speed is 250 feet per minute. The worm gears are carried in oil-tight cases, and the capstans, pawlplates and gear wheels are made of cast steel, the wheels having machine-cut teeth. The electrical equipment is also by Laurence, Scott & Electromotors Limited, the control being on the Ward-Leonard system.

The electrically-operated warping winches have been supplied by Clarke, Chapman & Co. Ltd., Gateshead-on-Tyne. Each is designed to exert a pull of 12 tons on the rope at a speed of 50 feet per minute, the light rope speed being 200 feet per minute. Each winch has a totally-enclosed watertight Clarke Chapman motor of 56 B.H.P. at 350 R.P.M., this motor driving warping ends 24 1/2 inches in diameter at the waist with 5/8-inch thick whelps cast on. The control system is on the makers' patent booster, which is a modification of the Ward-Leonard motor-generator control. Perhaps the most outstanding point about these warping winches is the great distance between the warping ends. In one case the distance is 20 feet between centers and in the other 31 feet between centers.

The navigation instruments include a Sperry Mark VIII compass. This, together with the necessary electrical auxiliaries, including an emergency battery to take care of failures of the ship's supply, is housed in a small compass room. This apparatus controls five repeater compasses. In view of the great convenience of the gyro compass as a means of taking bearings by the azimuth circle, this ship has no fewer than three bearing repeaters, one at each wing of the bridge and one on the center line on the flying bridge. As usual, there is a repeater compass for steering adjacent to the telemotor in the wheelhouse. As in so many modern ships, a repeater is also placed in the wireless room to assist the operator in obtaining radio bearings with the maximum accuracy. In this case it consists of the Sperry repeater attachment incorporated in the Marconi direction-finder. A Warwick pattern navigational light indicator has been supplied by Wm. McGeoch & Co. Ltd., Glasgow and Birmingham. An electric submerged log, by the Electric Submerged Log Company, is provided.

The steering gear fitted in the *Britannic* was supplied by Harland & Wolff Limited, and is of the electric-hydraulic type, consisting of four rams working on a double-ended tiller, all in accordance with the usual design of the makers. The main parts of the gear were manufactured by Harland & Wolff Limited, but the Hele-Shaw pumps were manufactured and supplied by John Hastie & Co. Ltd., Greenock. These pumps are two in number, and either is capable of steering the vessel at full speed. The gear is arranged so that both pumps may be run together or separately, and either unit may be instantaneously started up from the engine room. The gear is controlled by a telemotor of MacTaggart, Scott manufacure.

VENTILATION.

The passenger accommodation and public rooms throughout are mechanically ventilated and heated by Pleno-units, supplied by the Winsor Engineering Co. Ltd., Glasgow. The equipment consists of 42 Pleno-units, delivering 143,000 cubic feet of air per minute and arranged in suitable positions throughout the ship, care being taken to keep them clear of promenade and games decks. Each Pleno-unit heats and ventilates a section of the ship and each section is sub-divided, the temperature of the interior of the section being thermostatically-controlled and maintained within half a degree above or below the required temperature, which can also be varied at will from 0 degrees F. to 80 degrees F. Furthermore, the accommodation can be maintained at the latter temperature under the coldest Atlantic conditions and the Pleno-units are also designed to supply air at atmospheric temperature, as required. Instead of the usual directional type louvres, and because of their obtrusiveness, hit-and-miss louvres are used throughout and these are arranged in suitable positions.

Over 30 exhaust fans have been supplied by Davidson & Co. Ltd., Belfast, in connection with the ventilation system. These have been distributed

throughout the vessel so as to be as far away as possible from the supply fans. From such spaces as galleys, bakeries and pantries, which are important sources of wild heat, the air is extracted to give changes varying from 30 to 60 times an hour. Lavatories and bathrooms throughout the vessel have 30 changes of air per hour, and where possible the fans extracting air from these compartments take away the vitiated air which has been displaced from cabins into the passages adjoining the lavatories, &c. A feature of the crew space ventilation is mechanical exhaust ventilation in addition to warm air supply, so that it will be possible to maintain a constant renewal of air in these compartments in all weathers. Sirocco type exhaust fans also extract air from the cabin smoke room, long gallery, lounge, swimming pool, and all cabin entrances, so as to give changes of air at the rate of at least 15 an hour.

The whole of the hot air trunks, excluding those leading to the engine room, have been insulated by John Davies Insulating Co. Ltd., Bootle, Liverpool. In connection with this work Delta compressed cork has been used, which is manufactured by this company at their Bootle works. The trunks are finished with their special Pipco composition.

William McGeoch & Co. Ltd. have supplied large quantities of locks and hardware, and a considerable variety of electric fittings for various accommodation requirements, and other material in connection with lighting and bell circuits. They have also supplied some of the fittings for the public rooms in the third-class and tourist quarters. Roanoid Limited also supplied door furniture, finger plates, curtain rods, rings and brackets, electric fittings, &c., a mottled ivory finish being used for their products.

WIRELESS.

The *Britannic* is one of the first new ships to be affected by the regulations of the 1927 International Radiotelegraph Convention of Washington, and by one regulation in particular, which restricts the power of spark transmitters fitted after January 1, 1930, to 300 watts. The Marconi wireless installation is in a room at the after end of the chart room, and includes a valve transmitter and receiver, an emergency transmitter, and, as has been mentioned, a Marconi direction finder. Two motor lifeboats are also fitted with Marconi transmitters and receivers. The main transmitter is a valve installation for continuous and interrupted continuous wave signalling on 600 meters and 800 meters, and for continuous waves only on wavelengths between 1887 and 2725 meters. This transmitter will work with an aerial 335 feet in length, the horizontal span of which is 265 feet long and is 144 feet above the water line. There is a two-valve receiver, which is a new development in marine work, as it covers in one instrument the whole of the commercial wave-band of from 15 meters to 20,000 meters, for which two receivers were formerly used.

The Marconi direction finder in the wireless room will, of course, be used in conjunction with a Sperry gyro repeater, and the direction finder aerials are mounted on top of the chart room just forward of the Marconi room. These aerials are 7 feet high and are of the latest double loop type in which the actual aerial wires are totally enclosed in rigid metal tubes. Not only do they maintain the high standard of accuracy of the older types but the new aerial has the added advantage of occupying very little space and of being unobtrusive. It is probable that at a later date a short wave telegraph installation will be added to the present Marconi equipment, and this will give the *Britannic* a practically world-wide wireless range.

A complete system of Pul-syn-etic electric clocks has been installed by Gent & Co., Leicester. The installation comprises one Pul-syn-etic master

clock with 9-inch pilot dial and a patented system of automatic advance and retard whereby the complete installation of clocks may be either advanced or retarded as the ship proceeds either east or west. There are 59 secondary clocks of ornamental design to conform with particular decorations in the various positions.

A large installation of electric cabin bell indicators, comprising in all 23 indicators, is provided. Instead of the usual bell indicators there is installed outside each state room two small electric lights, one colored red for steward and the other green for stewardess. These are operated by similarly colored pushes in each state room. These lamps remain lit until the attending steward or stewardess operates the push which is fixed outside each state room door.

There is an elaborate system for repeating the band and gramophone music, by means of which the band playing in the cabin lounge is repeated to all other passenger accommodation throughout the ship. Alternatively, gramophone dance music may be transmitted. This equipment has also been supplied by Gent & Co.

Throughout the accommodation and in other parts of the vessel Foamite Firefoam fire extinguishers have been installed.

REFRIGERATING PLANT.

The refrigerating plant, which has been supplied and fitted on board by J. & E. Hall Limited, Dartford, deals with insulated cargo spaces in the Nos. 2 and 3 'tween decks, of a total insulated capacity of about 85,000 cubic feet, and a large set of ship's provision chambers situated in the 'tween decks forward of the auxiliary engine room. There are also a number of cold larders, cupboards in pantries, bars, &c., drinking-water coolers, and a large ice-making tank all connected to the brine system of the plant. The plant, which is placed on the lower deck at the forward end of the auxiliary engine room, comprises two large horizontal twin-compressor CO_2 machines, each driven by a variable-speed direct-coupled electric motor of 75 B.H.P. There are two sets of CO_2 condenser coils of copper tube, each contained in a circular cast-iron casing, and two sets of evaporator coils of steel tube, each contained in a circular steel casing. The evaporators, with the brine return tanks, regulators, &c., are placed in an insulated room. For circulating water through the CO_2 condensers there is a large vertical centrifugal pump driven by a variable speed electric motor. For brine circulation there are three Drysdale horizontal electrically-driven centrifugal pumps, with Laurence, Scott motors, also a special high-pressure centrifugal pump for delivering brine to the cupboards and water coolers high up in the ship. The brine system is arranged for the circulation of brine at freezing or chilling temperature as required in the different spaces for the carriage of various kinds of cargo. A brine heater is provided for the circulation of warm brine for thawing snow off the cooling pipes as required. The No. 2 'tween deck spaces are cooled by the circulation of cold air, two batteries of closely-nested brine pipes and two Sirocco fans, supplied by Davidson & Co. Ltd., being provided for this purpose. The No. 3 'tween deck cargo spaces and the insulated provision chambers are cooled by means of brine pipes arranged on the walls and overhead.

PROPELLING MACHINERY.

As is usual practice in large motorships, two separate engine rooms are provided, one for the main machinery and its auxiliaries, and one for the Diesel generating sets and other auxiliary plant. The propelling machinery

consists of two ten-cylinder double-acting four-stroke-cycle Diesel engines of Harland-B. & W. make, the cylinders being cooled by fresh water, while the pistons are oil cooled. Each engine is capable of developing 10,000 B.H.P. at 110 R.P.M., the cylinders being 840 millimeters in diameter by 1500 millimeters stroke. Although the engines are of the air-injection type, the injection air compressors are independently driven, a practice that has been followed in certain other large double-acting B. & W.-engined ships. Four independent injection air compressors, each driven by a four-cylinder trunk piston engine of Harland-B. & W. make, are located in the auxiliary engine room, and these also supply air to four reservoirs for manoeuvering purposes. Four 500-Kw. generators in the forward engine room, arranged to supply the necessary electric power for engine room and ship's purposes, are driven by six-cylinder trunk-piston engines of the same general type as the air compressor engines. One small steam-driven emergency compressor is also fitted.

WASTE-HEAT BOILERS.

A considerable proportion of the main and auxiliary engine exhaust gases is utilized for steam-raising purposes in waste-heat boilers. Five Clarkson thimble-tube type silencer boilers are installed on this ship, four of these boilers being capable of dealing with the exhaust gases from the two main engines. Each boiler has 500 square feet of heating surface, and is 5 feet 3 inches in diameter and about 11 feet in overall height, and is capable of producing a minimum of 2500 pounds of steam per hour at a working pressure of 100 pounds per square inch. This output can be exceeded in service, if desired, as each boiler is capable of recovering practically the whole of the available heat in the exhaust gases. Normally, however, a proportion of the gases will be by-passed direct to the funnel, as is often done in large waste-heat installations. No oil burners are supplied with these boilers, which are fitted with a Thermofeed feed-water regulator, supplied by Ronald Trist Limited, since with this arrangement the boilers will require practically no supervision when the ship is under way.

A fifth Clarkson boiler is installed for dealing with the exhaust gases from two Diesel generators. This small boiler has 250 square feet of heating surface and is capable of generating 1800/1900 pounds of steam per hour with a working pressure of 100 pounds per square inch. In addition to the five waste-heat boilers, two single-ended oil-fired Scotch boilers are also provided to supplement the steam supply, particularly in port. The direct-acting feed pumps used in connection with the boiler plant have been supplied by G. & J. Weir Limited. Pipes for carrying fuel oil, lubricating oil, steam piping, etc., have been supplied by Stewarts & Lloyds Limited, Glasgow. The distant reading water gauges for the boilers have been supplied by Kelvin, Bottomley & Baird Ltd.

Numerous motor-driven pumps have been supplied by Drysdale & Co. Ltd., Yoker, Glasgow. These include the main forced-lubrication pumps and the emergency bilge pump, the latter with Laurence, Scott motor. An evaporator, by G. & J. Weir Ltd., is provided. This is capable of evaporating 3000 gallons per day of 24 hours.

The main engine fuel oil is purified by a battery of seven Sharples No. 6 size centrifuges, these being of the totally-enclosed type. Each machine, which is driven by a 2½-horsepower electric motor, has a capacity of 300 gallons per hour. For treating the lubricating oil, two open type No. 6 size Sharples machines are installed on the port side of the main engine room, and two of the same size on the starboard side. Two similar machines are

installed in the auxiliary engine room. These machines are all British built, having been supplied by Super-Centrifugal Engineers Limited, London. It is interesting to note that the contract for supplying the lubricating oil to this notable vessel has been entrusted to W. B. Dick & Co. Ltd., their Ilo brand being used. The Diesel oil is supplied by the Asiatic Petroleum Company.

Ventilation of the main and auxiliary engine rooms is entrusted to six large motor-driven fans, which supply air to these compartments at atmospheric temperature. These fans are of the Sirocco type, and were supplied by Davidson & Co. Ltd., Belfast.

The electric engine-room telegraphs have been supplied by Siemens Bros. & Co. Ltd., Woolwich, who also supplied their electric distance thermometers and thermo-electric pyrometers to the vessel. Ford electric torsionmeters have also been supplied by the Woolwich firm.

There is an extensive installation of Graham telephones, communicating between the bridge and engine room. There are also connections with the engineers' accommodation and a direct installation between the captain and chief engineer. Graham instruments are also used for other parts of the ship, such as the purser's office and stewards' departments.—"Shipbuilding and Shipping Record," June 26, 1930.

THE TRANSATLANTIC LINER *LAFAYETTE*.

The Largest French Diesel-engine-driven Vessel. Quadruple-screw Double-acting M.A.N. Machinery of 18,000 S.H.P.

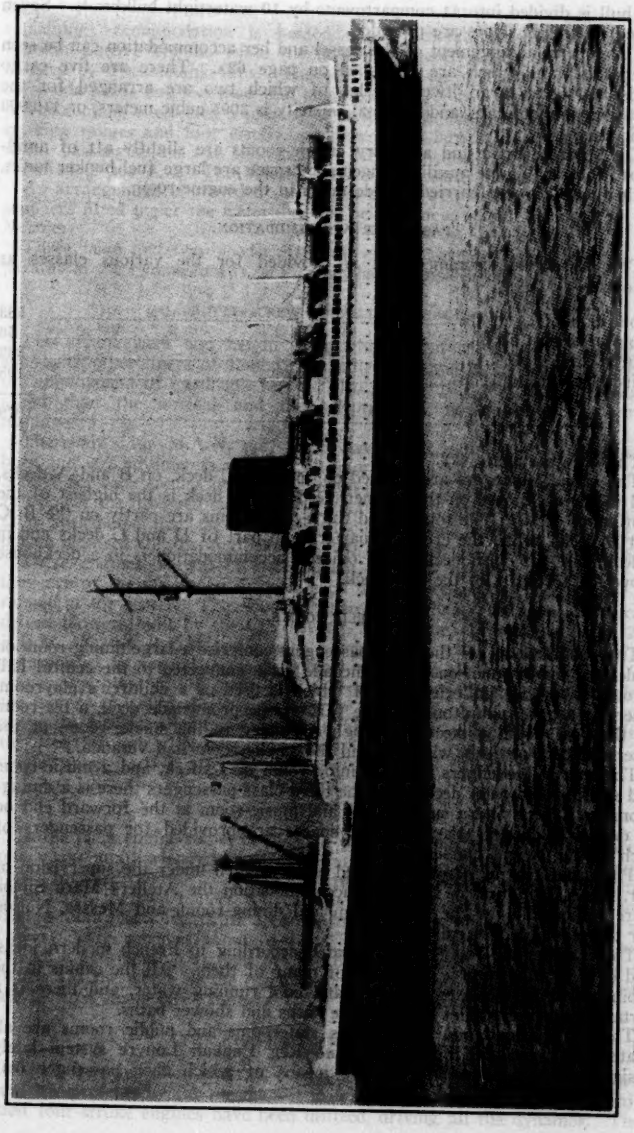
The *Lafayette*, which ran her acceptance trials last month and will sail on her maiden voyage from Havre on May 17th to New York, is by far the largest motor ship to fly the French flag, and is the third Diesel-engined vessel of the largest French shipping company, the Compagnie Générale Transatlantique. The *Lafayette* was built by the Soc. des Ch. et Ateliers de St. Nazaire-Penhoët, who constructed two of the four propelling engines, while the other two were built at the Augsburg works of the M.A.N. under the Dawes Plan.

The gross tonnage of the *Lafayette* has worked out somewhat greater than anticipated, being more than that of the *Saturnia* and *Vulcania*, so that as a completed motor ship the new vessel is second only in gross tonnage to the *Augustus*.

The main details of the *Lafayette* are as under:—

Length overall	183 meters or 610.24 feet
Length b.p.	175 meters or 574 feet
Moulded beam	23.6 meters or 77.34 feet
Depth	23.25 meters or 76.24 feet
Draught	9.25 meters or 30.3 feet
Displacement at this draught	25,600 tons
Gross tonnage	25,050
Deadweight capacity	9050 tons
Machinery power	16,000 to 18,000 S.H.P.
Service speed	17½ knots

The appearance of the ship is different from that of all other C.G.T. liners, as she has a large squat funnel, a single mast, a straight stem and cruiser stern. There are nine decks in all, of which five are continuous, and



THE "LAFAYETTE" (25,050 TONS GROSS), THE LARGEST MOTOR LINER TO SAIL UNDER THE FRENCH FLAG.

the hull is divided into 11 compartments by 10 watertight bulkheads. Seven Stone watertight doors are provided.

The general arrangement of the vessel and her accommodation can be seen from the plans which are published on page 62A. There are five cargo holds and four cargo 'tween decks, of which two are arranged for the transport of motorcars, and the total capacity is 3095 cubic meters, or 110,000 cubic feet.

The separate main and auxiliary engine-rooms are slightly aft of amidships; forward of the auxiliary machinery space are large fuel bunker tanks, while fuel oil is also carried in side tanks in the engine-room.

PASSENGER ACCOMMODATION.

The passenger accommodation is provided for the various classes as follows:—

Cabin class	583
Cabin or tourist class	56
Tourist	332
Third-class	108
Total	1079

The crew numbers 472.

The cabin passengers are berthed on part of C deck, on B and A decks, and in the deckhouse on the promenade deck. A deck is the highest of the continuous decks. The tourist and third-class cabins are partly on the B, C and D decks, while the crew is quartered on part of D and C decks and in a deckhouse aft on B deck. The officers' accommodation is in a deckhouse above the fore part of the boat deck.

PUBLIC ROOMS.

The public rooms for the cabin passengers comprise a large dining-room on C deck, a music room on the promenade deck, connected to the central hall through two wide galleries, one of which is used as a children's playroom. There is a gymnasium on the fore part of the promenade deck, a tea-room on the boat deck, connected through a gallery to the smoke-room, at the after end of that deck, while beyond the smoke-room is a veranda.

The tourist passengers have a dining-room on C deck, and a music-room and smoke-room on A deck. For the third-class passengers there is a dining-room on E deck and a smoke-room and music-room at the forward end on A deck. Open and glass-enclosed spaces are provided for passengers of each of the various classes.

The decorative work was designed and executed under the supervision of Mr. René Prou, and in the cabin accommodation the Ateliers Marc Simon were responsible for the music-room and dining-room, and Messrs. Nelson for the tea-room and galleries.

The main public rooms are decorated according to French modern ideas, and concealed lighting is a feature of many of them. All the cabins in the cabin and tourist classes have hot and cold running water, and there is a certain number of cabins with private baths and shower baths.

The first, second and mixed class staterooms and public rooms are all heated and ventilated on the Thermo Reg Punkah Louvre system lately designed by Thermotank, Ltd., by means of which each passenger may

control the temperature and quantity of air supplied to the cabin. The remaining accommodation is heated and ventilated on the Thermotank Punkah Louvre system, while the lavatories, pantries and galleys are exhausted by Thermotank fans. The quantity of air exhausted from the accommodation is not less than that supplied.

The cooking is effected wholly by electric means, and there are four electric cooking ranges and four electric roasters. The largest range, that for the cabin passengers, is not less than 24 feet 6 inches long, and the power required is 203 Kw. The baking oven is electrically heated.

An arrangement is provided for the discharge of waste water through the scuppers fitted under the waterline, this being a new system designed by Mr. Vernier.

Three main and two service lifts, all electric, are provided, and the apparatus in the gymnasium is on a somewhat extensive scale.

QUADRUPLE-SCREW PROPELLING MACHINERY.

The arrangement and height of machinery afford interesting comparisons with other liners of somewhat smaller size and equal engine power, in the employment of quadruple screws, also in the utilization of higher engine speed than the normal and the adoption of double-acting two-stroke machinery.

There are four M.A.N. units each having six cylinders 600 millimeters bore and 900 millimeters stroke. The normal output at 145 R.P.M. is 4000 S.H.P. for each engine, with a maximum of 4500 B.H.P. at 150 R.P.M. At full power the designed speed is 18¼ knots and the service speed 17½ knots. It is anticipated that the rate of fuel consumption per mile, including all power required for the auxiliaries, is 190 kilograms or about 420 pounds.

These engines, it will be noted, have standard cylinder dimensions, but they are of the air-injection class, each driving its own air compressor arranged at the center. The scavenging air is supplied from three electrically driven blowers built by the Cie. Electro-Mécanique, operating at from 2400 R.P.M. to 3000 R.P.M., each with a capacity of 1200 cubic meters of air per minute. In normal service two blowers provide all the air required. The scavenging blowers are located in a separate compartment, which encloses the motors as well as the blowers, at the after end of the main engine-room above the auxiliary pumps installed at this position. The air is drawn from the deck through a large suction trunk insulated to deaden the noise. The compartment itself is also well insulated.

ENGINE-ROOM LENGTHS.

In the main engine-room, besides the propelling machinery, are the various pumps and auxiliaries required for the main motors. In the auxiliary engine-room forward, and separated by a watertight bulkhead, are all the generating units, the two boilers and various ship's and other pumps. The main engine-room has a length of approximately 73 feet, while the auxiliary machinery space occupies about 64 feet. In these spaces, as previously mentioned, a considerable amount of fuel oil is carried in side tanks. The exhausts for the whole of the machinery, including the uptake for the boilers, are led to a single large funnel.

THE 3100-KW. GENERATING PLANT.

The generating plant is large, even for a liner of the size of the *Lafayette*, and it represents by far the biggest installation of its type, for airless-injection four-stroke engines have been utilized, driving all the dynamos. There

are five auxiliary engines, all of the M.A.N. type, three having six cylinders and developing 1050 B.H.P. These are coupled to 700 Kw. dynamos, and have cylinders 480 millimefers bore, with a piston stroke of 700 millimeters, the speed being 215 R.P.M. The two smaller engines are of 750 B.H.P., with six cylinders 425 millimeters bore and a piston stroke of 600 millimeters. They run at 250 R.P.M. and each is coupled to a 500 Kw. dynamo. Two of the larger engines were built at Augsburg and the remaining three units by the Société Générale de Constructions Mècaniques.

BOILERS.

The two cylindrical boilers are oil-fired on the Wallsend-Howden system, and operate at a pressure of 110 pounds to 115 pounds per square inch, the heating surface being 180 square meters, or about 1820 square feet. The Contra-flow condenser, feed pumps, evaporator and fresh-water distiller are located adjacent to the boilers.

In the main engine-room a large number of pumps are arranged at the after end. These include three 100-ton fuel oil pumps, two bilge pumps, each of 150 tons capacity, and three fresh-water cooling water pumps of 175 tons capacity. The last-named are of the centrifugal type and are for cooling the pistons of the main engines, sea water being employed for the cylinders and covers. There are three 500-ton sea-water pumps, two of 300 tons for water ballast and two 150-ton fire and deck-washing pumps. The 10-ton drinking-water pumps are also arranged at this point, while there are two oil separators.

Adjacent to the bulkhead between the main and auxiliary engine-rooms are the air reservoirs, these comprising eight of 2000 liters capacity each, utilized for starting and maneuvering, and four for injection air, maintained at a pressure of 75 kilograms per square centimeter, each of 2000 liters capacity.

The centrifugal purifying installation is arranged in the auxiliary engine-room and comprises nine Sharples centrifugal machines, five for dealing with the fuel oil and four to handle the lubricating oil. Also in the auxiliary engine-room space are three 100-ton centrifugal pumps for cooling the auxiliary engines, and another for the same purpose when in port with a capacity of 20 tons. There are three 100-ton sanitary pumps and two 100-ton oil transfer pumps, in addition to a lubricating oil pump. The two Reavell compressors are motor-driven, each being twin machines with a capacity of 1200 cubic meters of air per hour.

ON DECK.

Twenty-six lifeboats are carried, including 22 lifeboats and two whale boats supplied by Messrs. Hugh McLean, and two French-built motor lifeboats. There is a gyro-compass equipment and auto helmsman with a combined rudder recorder, also an electro-megaphone equipment for communication between the various parts of the vessel. The standard and steering compasses are of the Kelvite type, and there are two Kelvite heavy-duty motor sounding machines as well as Pneumercator draught indicators. A log of the Hughes type is provided and there is an electric Tyfon whistle. An echo sounding apparatus and recorder are installed, and there is a wireless apparatus with a radius of 1500 miles in the daytime and 3000 miles at night.

Hydro-electric steering gear of the Hele-Shaw design is provided, two pumps being fitted. With the complete gear in operation, the rudder can be

moved from hard aport to hard astarboard within 17 seconds at a speed of 18 knots.

For fire-extinguishing purposes the Lux Rich system is fitted in the cargo hold, while in the engine-room Foamite portable apparatus is installed.

For the ship's provisions there are 14 refrigerated rooms, the total volume being 867 cubic meters, or approximately 31,000 cubic feet. The refrigerating plant comprises two CO_2-type machines of the S.M.I.M. type, having a capacity of 110,000 frigories (negative calories) per hour.

CONSTANT-CURRENT SYSTEM OF TRANSMISSION.

Perhaps the most notable feature of the auxiliary installation is in the employment of the constant-current system for driving the deck machinery. This is operated on the Gilbert Austin system, which has been described and illustrated in this journal.* It may be recalled that the principle of this arrangement is the employment of constant current instead of constant voltage, so that fuses are eliminated, and the claim is made that the control gear for the winches is simplified, resistances are not required and overloading is impossible. The load can be held stationary in the air as long as required without using a brake, while when a load is being lowered from a winch, power is returned to the line, thus providing a regenerative effect. The *Lafayette* is the first French vessel in which the constant-current system has been employed.

The deck machinery consists of a 60-ton double windlass with two motors, five 20-ton capstans, five 5-ton winches, three of these being fitted with extra large drums of 16 to 20 tons lift, nine 3-ton winches and two 1½-ton winches. The main contractors for the installation were Gilbert Austin, Ltd., Cathcart, and the sub-contractors for the mechanical portion of the gear Messrs. Napier Bros., Ltd., for the windlass, and the Soc. Anonyme des Ateliers Duchesne et des Ateliers Bossiere, Le Havre, for the capstans and winches.

The windlass is in two complete units, each working independently, so arranged that one machine may be made to operate the other. The arrangement thus gives the equivalent of four independent capstans, since the windlass warping drums are each independently controlled by their motors.

At the after end of the ship the capstans are placed one on each side and one amidships. The windlass and capstan controllers are arranged for 209 speed steps in either direction. The controller, with 11 steps in each direction, has full command of the torque and therefore of the speed under any condition, but the accelerator with 19 steps acts as a vernier adjustment on each speed step, thus giving a total of 209 steps forward and reverse.

The speed range, as compared with capstans on British ships, is somewhat unusual. The full load of the capstans is 40 feet per minute and the light rope speed 240 feet per minute. This large range is performed by the motor without change gear, and the full load range of zero to 40 feet is also included in the range of the capstan. When tested, while the ship was being manoeuvered from the shipyard, even with seven motors in use, each of the 150 H.P., the average load was only 25 Kw. and the maximum 80 Kw.

The 5-ton winches can run at any speed from zero to 400 feet per minute. When heavy loads are to be lifted, the motor is uncoupled from the worm and connected to a spur reduction gear without increasing the ratio of reduction.

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*"The British Motor Ship," July 1924, pages 122 and 123.

For the supply of constant current to the system, two motor generators, each of 150 Kw., are provided. They are placed in the emergency dynamo room, together with the constant-current control board and the starting panels for the motor generators.—"British Motorship," May, 1930.

THE OPERATION OF THE *LAFAYETTE*

Machinery Details and Particulars of the M.A.N.-Type Main and Auxiliary Engines at the Outset of the Maiden Voyage.

Before the largest French motor liner, the *Lafayette*, departed from Le Havre on her maiden voyage to New York we carried out a thorough inspection of the vessel; also, at the invitation of the owners, we were present in the engine-room during the entire series of manoeuvers leaving Havre and arriving at Plymouth. Further, for a considerable time we noted the operation of the four main engines at the normal service speed and obtained the necessary details relative to the auxiliary Diesel machinery, four of the five dynamo engines being in requisition. The main engine designed output in this quadruple-screw liner is 16,000 B.H.P., and the aggregate power of the auxiliary Diesel motors driving the dynamos is 4650 B.H.P.

As a full description of the *Lafayette*, together with complete engine-room and ship plans, has already appeared in this journal*, the following account will be largely based on the results obtained from her first trip in service, so far as the Le Havre-Plymouth run is concerned. A stiff breeze, accompanied by one or two sharp squalls, sprang up in the early morning before the vessel reached Plymouth, and although the weather was scarcely to be termed bad, it gave an indication that the new ship might be expected to exhibit satisfactory behavior in relatively heavy seas. If this proves to be the case, her accommodation should be well filled, for the passenger cabins and public rooms which we inspected are most attractive in their design and arrangement.

Modern French architecture is, in certain respects, inclined to a measure of solidity in appearance. This is perhaps reflected in the *Lafayette's* accommodation; yet there is brightness and comfort throughout, the general furnishing and schemes of decoration defeating, as it were, any attempt at stiffness brought about by the main design of the apartments. The lighting schemes are good, partly concealed and partly open in their effects, while the provision of private shower baths, with hot and cold water, in many of the cabins, is bound to be appreciated.

THE DECK EQUIPMENT.

Owned by the Cie. Générale Transatlantique, the *Lafayette* has a gross register of 25,050 tons and is 610 feet 3 inches overall, the beam being 77 feet 4 inches and the depth 76 feet 3 inches. On a displacement of 25,600 tons the vessel draws about 30 feet 4 inches of water. She has accommodation for 1079 passengers. All the auxiliary machinery is electrically operated, and the deck equipment is arranged to work on the Gilbert Austin constant-current system. The principle was explained in the article on the *Lafayette* to which reference has been made.

The Napier anchor windlass forward is in two sections, with twin electric motors on deck, Austin control gear being fitted. The cargo winches, similarly controlled, have been supplied by the Ateliers Bossiere, Le Havre.

* "The British Motor Ship," May, 1930, pp. 62a, 63, 64, 65, 66, 67.

In the steering-machinery compartment aft is the electro-hydraulic steering gear, with four rams arranged athwartships, two pumps and motors being fitted. The pumps are of the Hele-Shaw type, supplied by the Paris licensees. Changeover valves are provided to order to use either the port or starboard pump, or both. Each of the electric motors, manufactured by the Cie. Française Thomson Houston, Paris, is rated at 32-42 Kw., and is of the compound-wound type, taking current at 220 volts. In the steering compartment are also arranged the motors for the after capstans (port and starboard), while above, on deck, is a central capstan with the motor complete.

On trials the *Lafayette* achieved a maximum speed of 18.3 knots with the engines turning at 154 R.P.M. The service speed of the vessel is approximately 17.5 knots, and the total output required from the four engines is then about 16,200 I.H.P. It is arranged to run the two inner main engines at 148 R.P.M. and the wing units at 150 R.P.M. or 151 R.P.M., to avoid synchronizing and the corresponding risk of vibration. We doubt, however, if any question of vibration could seriously be raised in the *Lafayette*, for at the normal full speed of the ship it is—at any rate, in some parts of the vessel—difficult to realize that the engines are in operation. No doubt the enthusiastic seeker after vibration may find it somewhere in the accommodation, but he would certainly be making a fruitless search in most parts of the public rooms. The fact that the engines are in motion can be detected in some of the cabins, but we were quite unable to decide whether the main or auxiliary machinery was responsible for the minor effects produced. In the first-class dining saloon, to take another example, one might almost imagine that—at full speed, be it emphasized—the apartment was on dry land, given reasonable weather conditions.

MAIN ENGINE CONDITIONS AT SEA.

In the following tables, as regards the main propelling engines, the revolutions are given in accordance with the counter readings at the particular moment of observation and may not agree exactly with an average result for a set period. We give all the figures, however, precisely as they were noted at the time, and no allowances whatever have been made for possible variations.

	Stbd. wing eng.	Stbd. inner eng.	Port inner eng.	Port wing eng.
Revs. per min.	150	148	151	152
Blast-air cpr., L.P. stage	35.5 lb. per sq. in.	30.5 lb. per sq. in.	32.5 lb. per sq. in.	31.5 lb. per sq. in.
Intermediate stage	177.5 lb. per sq. in.	162.5 lb. per sq. in.	170 lb. per sq. in.	170 lb. per sq. in.
Piston-cooling pressure	39 lb. per sq. in.	39.5 lb. per sq. in.	38.5 lb. per sq. in.	39.5 lb. per sq. in.
Jacket-cooling pressure	21 lb. per sq. in.	20 lb. per sq. in.	21 lb. per sq. in.	20 lb. per sq. in.
Forced lub.-oil pressure	29 lb. per sq. in.	25.5 lb. per sq. in.	31 lb. per sq. in.	30.5 lb. per sq. in.
Blast-air pressure	920 lb. per sq. in.	920 lb. per sq. in.	910 lb. per sq. in.	910 lb. per sq. in.
Fuel wheel setting	90-120	64-120	55-120	84-120

Turbo-blower scavenging air pressure 1.7 lb. per sq. in.

It will be observed that the lowest revolutions are 148 per minute (starboard inner engine), and the highest 152 per minute (port wing engine). The pressures are conversions from kilogram per square centimeter to the nearest 0.5 pound per square inch.

The blast-air pressure is maintained by hand regulation, so that at different speeds the figure varies to suit the conditions. It is found that in the case of these engines the pressures quoted suit the revolutions to which they apply.

It will be noted that at about 150 R.P.M. the blast-air pressures were lower than the specified figure, but this was no doubt due to special circumstances. As regards the auxiliary engine-room conditions, the table gives the running results of all engines in operation at the time, viz., all three 700 Kw. sets and one of the two 500 Kw. units.

The load on all four machines was considerably less than the rated capacity, and there was ample reserve in hand preparatory to starting the two auxiliary manoeuvering air compressors, each of which may take approximately 600 ampères at full load. These compressors are of the Reavell four-stage type, each coupled to an Allen compound-wound electric motor with a continuous rating of 160 B.H.P. at 72 degrees F., the speed being 450 R.P.M. In addition, there is a small machine driven by an Allen motor of 8 H.P., the speed being the same as with the larger compressors.

BLAST-AIR PRESSURES FOR DIFFERENT SPEEDS.

Normal Speed.	*Revs. per min.*	*Blast-air press. lb. per sq. in.*
Slow	60	710
Half	110	850
Full	145	990

At the forward end of the auxiliary engine-room is the main switchboard, which extends nearly the whole breadth of the ship. It is a well-finished installation built by the Soc. Anon. Merlin et Gerin, Grenoble. The refrigerating plant is installed aft of the main engine-room and comprises two electrically driven CO_2 compressors. Each compressor is driven by an 80 H.P. Schneider motor taking current at 200-220 volts. The equipment includes two rotary brine pumps.

CENTRIFUGAL PURIFYING INSTALLATION.

There are two batteries of Sharples centrifugal purifiers. Five of these machines are installed in the auxiliary engine-room on the starboard side and deal with the fuel for all the Diesel engines, main and auxiliary. They are of the enclosed type and on each machine is a record showing the time of its last cleaning out. Four machines, reserved for purifying the lubricating oil, are fitted in the main engine-room. These are also of the Sharples type, so that the complete installation forms a battery of nine purifiers—one of the largest of its type afloat.

A precise estimate of the fuel consumption for all purposes is difficult to form at this early stage in the ship's service, but, as a rough idea of the expenditure, it may be assumed that the main engines will account for about 60 tons of fuel per 24 hours, while another 10 tons or so should be sufficient to cover the auxiliary engine requirements. The fuel capacity of the vessel

is large enough to allow a certain quantity of oil to be discharged at the home port on each round voyage. There are two oil-fired boilers installed in the auxiliary engine-room, and it is somewhat surprising to find that no advantage is taken of the exhaust-gas heat from the auxiliary engines, which are of the four-stroke airless-injection design.

THE MAIN ENGINE CONTROLS.

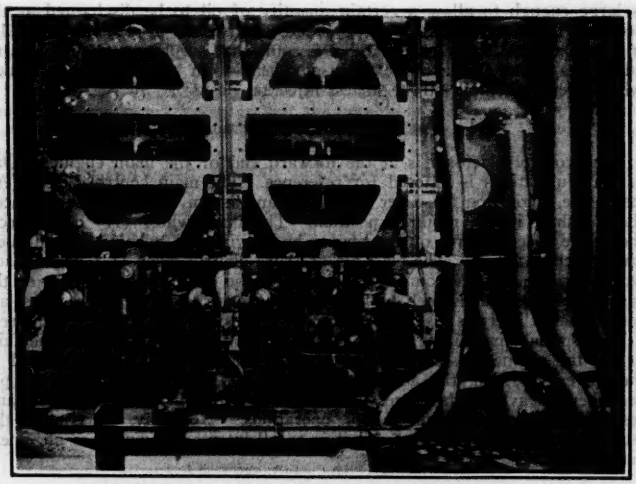
Each control platform is at the forward end of the main engines. One of the illustrations is lettered for reference. The main control wheel (A) starts the engine on air after the ahead-and-astern lever (B) has placed the camshaft in the required position. The main wheel (A) also affects the entire speed control of the engine by varying the opening of the fuel-pump suction valves. The blast-air pressure control lever (C) has already received reference. A pointer (D) indicates the direction of rotation of the engine, while a handwheel (E) regulates the air suction to the blast-air compressor. There is also a lever for giving extra starting air, if required.

This special air-starting lever is an additional fitting; assuming the failure of the engine to start on fuel when the main wheel (A) is turned to the correct position, the additional air lever gives an independent supply and thereby avoids the necessity of turning the main wheel back and restarting the whole sequence of operations. Normally, of course, the lever in question is not used and it is to all intents and purposes an emergency fitting.

AUXILIARY ENGINE-ROOM CONDITIONS.

	Stbd. 700 Kw. Machine	Center 700 Kw. Machine	Port 700 Kw. Machine	Port 500 Kw. Machine
Speed	213 R.P.M.	213 R.P.M.	215 R.P.M.	250 R.P.M.
Lubricating oil pressure (before filter)	49.5 lb. per sq. in.	56 lb. per sq. in.	53 lb. per sq. in.	74.5 lb. per sq. in.
Lubricating oil pressure (after filter)	42.5 lb. per sq. in.	49.5 lb. per sq. in.	46 lb. per sq. in.	67 lb. per sq. in.
Lubricating oil pressure (after cooler)	46 lb. per sq. in.	50 lb. per sq. in.	44.5 lb. per sq. in.	49.5 lb. per sq. in.
Forced lubricating oil to bearings	27 lb. per sq. in.	28.5 lb. per sq. in.	27 lb. per sq. in.	27 lb. per sq. in.
Piston-cooling pressure	41 lb. per sq. in.	42.5 lb. per sq. in.	42.5 lb. per sq. in.	....
Jacket water-cooling pressure	25 lb. per sq. in.	27.5 lb. per sq. in.	26.5 lb. per sq. in.	28.5 lb. per sq. in.
Load on dynamo (amps.)	2250	1950	2000	1650
Approximate Kw.	495	430	440	360

Each main engine has six cylinders, 600 millimeter bore and 900 millimeter stroke. The blast-air compressor is arranged centrally and is driven from the crankshaft.



THIS VIEW CLEARLY SHOWS THE ARRANGEMENT OF THE SCAVENGING—AIR AND EXHAUST PORTS IN THE M.A.N. ENGINE.

VIEW LOOKING ACROSS THE TOP OF TWO OF THE FOUR MAIN ENGINES.

THE SCAVENGING AIR BLOWERS.

Scavenging air is supplied by three electrically driven blowers, two being in action while the third is a stand-by. These blowers, with their electric motors, were supplied by the Cie. Electro-Mécanique, and each has a capacity of 1200 cubic meters of air per minute. The maximum blower speed is 3000 R.P.M. At the forward end of each scavenging air trunk is a diaphragm plate for safety purposes; the plate is sufficiently thin to burst in the event of an excessive rise of pressure in the pipe. Suitable orifices around the end circumference of the pipe are arranged to discharge the excess pressure to the atmosphere and prevent damage to the engine.

The fuel cams on the camshaft are tapered off to the diameter of the shaft and no provision is otherwise made for lifting the valve lever rollers when the camshaft moves fore and aft for reversing. Normally the air-starting rollers are clear of their respective cams until the moment they are needed. It will be seen, therefore, that the whole reversing system and control gear are simplified. The efficiency of manoeuvering was well demonstrated by the ship's engineers and numerous manoeuvers were carried out with obvious efficiency and speed.

COVER DESIGN.

It is necessary to remark briefly on the design of the cylinder covers. The engines in the *Lafayette* are of the standard M.A.N. double-acting two-stroke type and need no extended description. The upper and lower covers are mainly identical, inasmuch as there is an orifice for the fuel valve in the top cover (starting air being supplied to the underside of the pistons only) and a hole for the piston rod and stuffing box in the bottom. Actually, these covers are not interchangeable, as the bottom orifice for the piston rod is somewhat larger than that for the fuel valve at the top.

There are eight cams per cylinder. Two are for the top fuel valve (one for ahead and the other for astern) which is operated by a vertical shaft. Two are required for the front fuel valve in the lower cylinder cover and two more for the fuel valve at the back, operated by a horizontal shaft. The remaining pair of cams are for air starting at the bottom.

STAINLESS STEEL PISTON-COOLING PIPES.

The fresh-water piston-cooling pipes are of stainless steel with ordinary leather packing, which, we are informed, is found very efficient. The glands for the telescopic pipes are arranged in small recesses at the side of the engine, entirely outside the crank chamber and readily visible to the engineer. The cylinder jackets are cooled by salt water. The port and starboard inner main engines are Penhöet-M.A.N. units, while the wing engines were built by the M.A.N. at Augsburg. Usually, manoeuvering in and out of harbor is effected by the wing engines alone and the two inner units are mostly reserved for full speed.

The Cie. Générale Transatlantique have added a fine quadruple-screw liner to their fleet, and, together with the builders of the hull and machinery, will be unreservedly congratulated on the ship. We were interested to watch her extremely efficient performance on the trip to which we have referred and fully expect that this efficiency will be characteristic of the vessel throughout her service.—"British Motorship," June, 1930.

A NEW HIGH-PRESSURE TURBINE AND BOILER.

Light High-speed Unit Installed in a German Patrol Boat.

A unique type of high-speed turbine and high-pressure boiler has been installed in the patrol boat *Hindenburg* for the German customs authorities; and the trials of the plant, which is the first of its kind, were successfully completed at the end of March.

The vessel, which is illustrated in Figure 1, has a length of 102 feet, a breadth of 14 feet 6 inches, and a draught of 4 feet, with a displacement of 60 tons. For a number of years it was driven by a petrol engine; and when

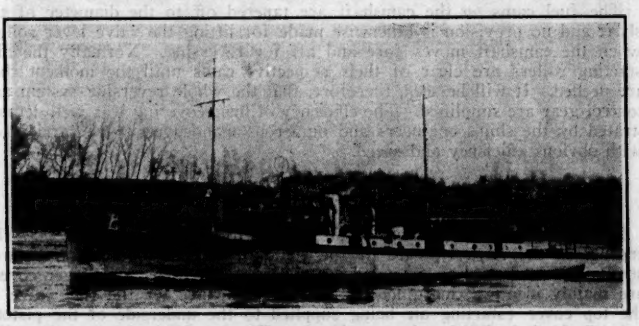


Fig. 1.—The German Customs Patrol Boat "Hindenburg."

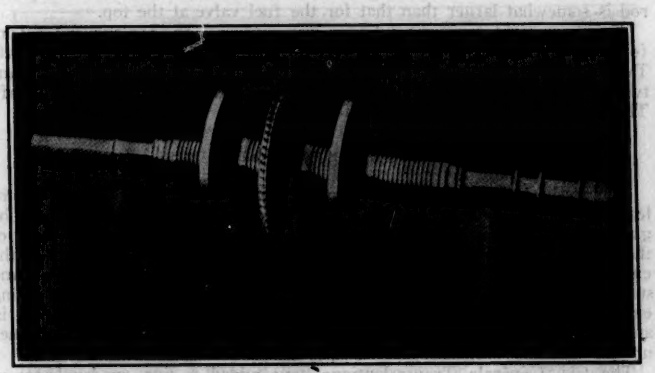


Fig. 4.—The High-pressure Rotor.

this was worn out and it was necessary to replace it by a more economical engine, it was thought desirable to install a high-pressure, high-speed turbine, in order to gain experience with this type of plant before fitting similar units in two new patrol boats for the same service.

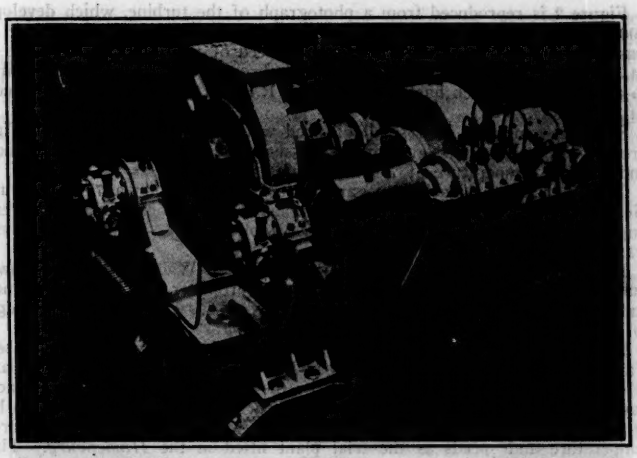


FIG. 2.—THE TWO-STAGE HIGH-SPEED TURBINE AND REDUCTION GEAR.

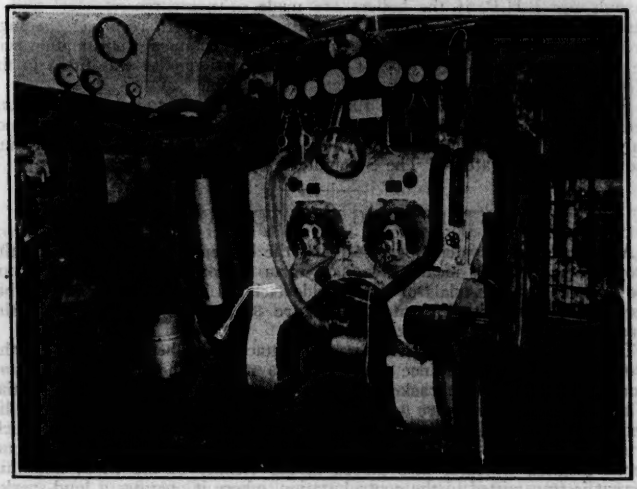


FIG. 3.—THE OIL-FIRED HIGH-PRESSURE WATER-TUBE BOILER.

Figure 2 is reproduced from a photograph of the turbine, which develops 300 B.H.P. in two stages. The high-pressure unit, illustrated in Figure 4, has a speed of 21,000 R.P.M., while the speed of the low-pressure stage is 14,300 R.P.M. The power is transmitted through double-reduction gearing to the propeller, which has a speed of 500 R.P.M., giving the vessel a speed of 13.5 knots.

Steam is generated in a small oil-fired water-tube boiler, shown in Figure 3. The boiler is designed for a pressure of 350 pounds per square inch and a steam temperature of 790 degrees F.

On the trials, the machinery gave entire satisfaction on a 7½-hour run. Neither the engine nor the gearing made any noise—a fact which is of great importance in a vessel of this class. No vibration was felt, and reversing was accomplished in a very short time.

The design of the machinery is due to Dr. Rudolf Wagner, the well-known marine engineer and inventor of the contra propeller and of the Wagner rudder. For some time Dr. Wagner has devoted his energy and knowledge to the development of high-pressure engineering.

The conversion of the vessel was carried out at the yard of the Fr. Lürssen Yacht-u. Bootswerft, of Vegesack, near Bremen. The two new patrol boats referred to are also being constructed by the same firm, and these will each have a length of 95 feet and a speed of about 29 knots. They will each be fitted with a turbine of 1600 B.H.P., which will have the same pressure, temperature and speeds as the trial plant fitted in the *Hindenburg*. The weight of the engines, including auxiliaries, boilers, shafting and propellers, will be about 20 pounds per B.H.P.—a remarkably low figure for so small a plant. The Wagner Hochdruck Dampfturbinen A.G., of Hamburg, the designers of the machinery, claim that this figure will be reduced to 13.2 pounds per B.H.P. if the boilers are made of non-corrosive steel and if certain other novel developments are introduced. As a result of the high pressure employed, it is expected that a very economical consumption will be obtained with this new type of plant. No torsionmeter was fitted in the *Hindenburg*, and consequently no data are available.

For the new patrol boats, the fuel consumption is not to exceed 0.78 pound per B.H.P., including that taken by the auxiliaries. This economy is not only due to the high pressure, but also to an efficient system of preheating the air and feed water.—"The Shipbuilder," June, 1930.

AN INTERESTING NEW TORSIONMETER.

A very accurate system for measuring the horsepower transmitted by the propeller shafting by means of a new form of torsionmeter was tried for the first time on the trials of the Norddeutscher Lloyd liner *Bremen*. This instrument, which is illustrated in Figure 3, has been developed by the Hamburg firm of K. Maihak.

Two wrought-iron rings, with brackets attached to them, are fixed to the propeller shaft at a distance apart of 0.425 meter (about 17 inches); and from the points of the brackets there is stretched, in a direction tangential to the shafting, a steel wire 0.150 meter (about 6 inches) long and 0.3 millimeter diameter, while above the wire there is an electro-magnet by which it is vibrated. This vibration generates an alternating current in a second electro-magnet, which is also placed in the vicinity of the wire; and this current is transmitted to the control station, where it operates a loud-speaking telephone. At this station there is a second steel wire, the tension of which may be regulated by a micrometer screw, and this wire is also provided with a similar electro-transmitting system, which is in turn connected to another loud-speaking telephone.

When it is required to take a reading, the wire on the shaft is put under a known tension with the shaft at rest, so that the sound which it emits has a definite pitch which may be heard at the control station. The wire on the instrument at this station is also stretched so that it gives the same pitch. When the shaft is transmitting power, the wire is further stretched, and the pitch of the note produced will rise. In order to ascertain the amount of elongation, the wire at the control station is stretched until the note which it produces is of the same pitch.

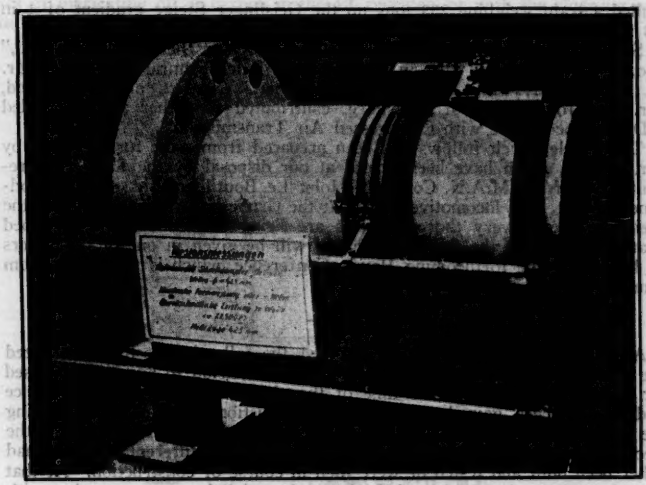


FIG. 3.—THE NEW TORSIONMETER USED ON THE TRIALS OF THE "BREMEN."

It is not a difficult matter and does not require a musical ear to tune the two notes; for if there is the slightest difference in the pitch of the notes, due to a difference in the number of vibrations per second of the wires, a gradual rise and fall in the strength of the note, which can easily be detected by the human ear, will be produced. Only when the sound is entirely regular and even does the number of vibrations per second coincide.

The instrument is divided into 400 divisions, each corresponding to a certain elongation. On the instrument used in the *Bremen*, each division was equivalent to a rise in power of 100 H.P., and on the trials about 22,500 B.H.P. was measured by the instrument on each line of shafting. For small engines, the instrument may be so adjusted that one division of sound difference corresponds to only 1 B.H.P. The telephone which takes up the sound at the control station can be switched on to any of the recording instruments if there is a number of these on different shafts. As it takes only a very short time to adjust the control instrument, several readings may be taken within a few minutes. The instrument is short and compact, and is extremely sensitive and accurate. Since the trials of the *Bremen*, it has been tried in a number of other ships, and it is now developed so that it will work continuously.—"The Shipbuilder," June, 1930.

1200 B.H.P. DIESEL-COMPRESSED AIR LOCOMOTIVE FOR THE GERMAN STATE RAILWAYS.

In our annual review of locomotive progress during 1929, which appeared in "The Engineer" for January 3rd, 1930, brief mention was made of the new Diesel-compressed air locomotive, which has been designed and built by the Maschinenfabrik Augsburg Nürnberg A.G. and the Maschinenfabrik Esslingen for the German State Railway Co. It may be recalled that in the official tests carried out in the Stuttgart district in November last, the engine pulled a train weighing 233 tons up the Geislinger Steige gradient of 1 in 43 mostly on curves, at an average speed of over 20 kilometers per hour.

Some particulars of this locomotive were given in the "V.D.I. Journal," No. 10, of March 8th last, in an article by Reichsbahnrat Witte and Mr. R. P. Wagner, while in the same "Journal," No. 12, of March 22nd, Dr. Ing. Jos. Geiger, of Augsburg, contributed a further article entitled "Diesel Locomotive with Compressed Air Transmission."

The article which follows has been prepared from some further notes by Dr. Geiger, which have been placed at our disposal by the London representatives of the M.A.N. Company—John Le Boutillier, Ltd., of 13, Roodlane, E.C. 3. The locomotive is now at the Grunewald railway shops of the Reichsbahn, and very thorough tests are being carried out. It is hoped that an account of the test performance will be given in one of the papers to be read at the forthcoming Power Conference, which is to be held from June 16th to the 25th in Berlin.

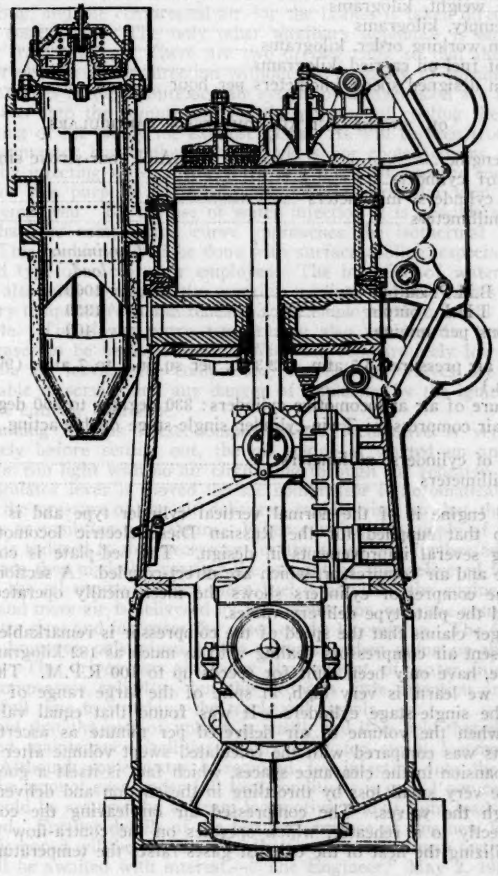
At the time the locomotive was ordered, the following position influenced the builders and the railway officials in their decision to adopt compressed air transmission. Up to the time of the placing of the order experience had already been gained with internal combustion locomotives embodying the direct drive, the geared drive, the fluid transmission drive, and the electric drive, while locomotives embodying these different systems had either been built and tested or were then in course of construction. At that time, however, a locomotive of the internal combustion type operating with compressed air transmission had not been built. On many sides the opinion prevailed that such a method of power transmission was valuable on several grounds. Thus, the hyperbolic curve of tractive power for compressed air working approached very closely that for steam and fitted in well. Again, by utilizing compressed air, difficult manoeuvering operations were avoided, and the driver had only to attend to the regulation of the supply of fuel to the oil engine. Another advantage of the compressed air system, it was held, was that the driver would have to deal with essential parts of the locomotive already familiar to him in steam practice.

On the other hand, there were those who foresaw difficulties, and did not hesitate to prophesy failure. It was said that the efficiency of the compressed air system of power transmission was lower than that of any other system, and it was also suggested that explosions arising from oil in the compressed air might take place, while stoppages could occur owing to the formation of ice on the cylinders in which the air was to be expanded.

Such statements were made even after actual experience with the locomotive showed that such happenings did not occur. We are informed that, in fact, the efficiency of transmission with the oil-compressed air system has been demonstrated to be higher than with the oil-electric system, enabling a 12 per cent saving on the fuel used per ton-kilometer to be recorded at a

common speed of 60 kilometers per hour. There have been no explosions of oily vapor, even although in a works test it was sought to produce such explosions artificially by the introduction of electric sparks.

During the long series of trials which have been carried out, there was at no time any signs of ice formation on the cylinders of the locomotive. On the contrary, when the hand was placed in the air at the exhaust outlet it was quickly withdrawn, since the air, after being expanded to close upon atmospheric pressure, still retained a temperature of about 100 degrees C. No difficulties have been experienced with the deposit of dirt from particles of lubricating oil or carbon in the air reheater, or with the clogging of the air compressor valves from like causes.



AIR-COMPRESSOR.

GENERAL PARTICULARS.

In the following table the general particulars of the locomotive and the leading dimensions of the oil engine and air compressor are given.

LOCOMOTIVE PARTICULARS.

Type	4—6—4
Tractive effort at driving wheel rims, kilograms	12,000
Diameter of driving wheels, millimeters	1600
Diameter of locomotive cylinders, millimeters	700
Stroke of locomotive cylinders, millimeters	700
Adhesive weight, kilograms	54,600
Weight empty, kilograms	118,600
Weight in working order, kilograms	124,000
Weight of fuel oil carried, kilograms	2,000
Maximum designed speed, kilometers per hour	80

OIL ENGINE AND AIR COMPRESSOR PARTICULARS.

Type of engine	M.A.N. four-stroke single-acting
Number of cylinders	Six
Bore of cylinders, millimeters	450
Stroke, millimeters	420

	Continuous load.	*Short overload.*
Designed B.H.P. output	1000	1200
Designed I.H.P. output	1350	1630
Revolutions per minute	400	450

Working air pressure: 6.5 atm. (92.3 lb. per sq. in.) to 7 atm. (99.4 lb. per sq. in.)
Temperature of air at locomotive cylinders: 330 degrees to 360 degrees C.
Type of air compressor: Twin-cylinder single-stage double-acting.

Diameter of cylinders, millimeters	640
Stroke, millimeters	350

The oil engine is of the normal vertical cylinder type and is generally similar to that supplied for the Russian Diesel-electric locomotives, but embodying several improvements in design. The bed-plate is common to the engine and air compressor, which are direct-coupled. A section through one of the compressor cylinders shows the mechanically operated suction valves and the plate-type delivery valves.

Dr. Geiger claims that the speed of the compressor is remarkable since up to the present air compressors dealing with as much as 182 kilograms of air per minute, have only been built for speeds up to 100 R.P.M. The overall efficiency, we learn, is very high, in spite of the large range of compression, in the single-stage cylinders. It was found that equal values were obtained when the volume of air delivered per minute as ascertained by nozzle tests was compared with the calculated swept volume after allowing for re-expansion in the clearance spaces, which fact is itself a good indication of the very small loss by throttling in the suction and delivery of the air through the valves. The compressed air on leaving the compressor passes directly to a reheater, which operates on the contra-flow principle, and by utilizing the heat of the exhaust gases raises the temperature of the

air to 360 degrees C. From the reheater the air passes through the regulator control valve to the two cylinders of the locomotive in which it is expanded to practically atmospheric pressure.

It will be noted that laminated coolers of the fully welded type are arranged at both ends of the locomotive. These coolers serve the circulating water system for the engine jackets and bearings, the piston cooling oil circuits, and the lubricating oil circuits. The compressor is cooled by the direct injection of water, which will be referred to later.

Under normal conditions, the natural draught through the coolers is sufficient, but electrically driven fans are provided to meet exceptional cases.

A small direct-driven dynamo provides current for the fan motors and for train lighting, and the compressed air for the brakes is taken directly from the main compressor. The only other auxiliary appliance is an oil-fired boiler for train heating. There are two driving cabs, so that the locomotive can proceed in either direction without the necessity of turning.

Two essentials of the compressed air system of transmission are the injection of water into the compressor for the purpose of cooling the air, and the reheating of the air by the exhaust gases. As will be seen from Figure the air compressor has neither cooled jackets nor cooled covers. The air is cooled by injecting water in the form of a fine mist into the compression space. For this purpose a special pump with a particular kind of spray nozzle is employed. By the use of water injection it is possible so to cool the air that the compression curve approaches the isothermal line very closely. This could not well be done with surface cooling, especially in the high-speed type of compressor employed. The injection of water into the cylinders also adds weight to the working medium and allows a very much lower entry temperature at the reheater to be employed than would otherwise be possible. This lower entry temperature also enables the heat of the exhaust gases to be better utilized. With the comparatively low temperature of the air leaving the compressor, the delivery valves are rendered more reliable in service and any danger of explosion by the ignition of oil particles is entirely overcome.

The running of this Diesel-compressed air locomotive is very simple. Immediately before setting out, the oil engine is started up and the air compressor run light with no air circulating through the reheater. As soon as the regulator lever is moved the air compressor is automatically put on load and the pressure of the air delivered rapidly rises so that the locomotive starts as easily as with steam. When running the supply of air to the cylinders is reduced to that necessary for the speed and further adjustments in speed are then made by varying the amount of fuel delivered to the cylinders of the oil engine. When more fuel is admitted the engine increases its speed and more air is delivered by the compressor, with the result that the air pressure rises and increases the speed of the locomotive. The full supply of air is, however, only given to the cylinders when going up the steepest gradients. The driver can also vary the amount of water injected into the air compressor cylinders, but this requires little attention and he can give his full time to watching the track and signals.

It is said that the cost of upkeep is small, and it is held that the normal repair and maintenance facilities as arranged for steam locomotives will suffice. Although some water is required for injecting into the air compressor the amount even under full load conditions is only 1 per cent of that which would be required by a steam locomotive, and at light loads no water is required. Further details of the test performance of this new locomotive, which it is hoped will be published by the Grunewald railway testing department, will be awaited with interest.—"The Engineer," May 2, 1930.

HULL DAMAGE REPAIRS TO THE U. S. LINER *LEVIATHAN*.

By Capt. C. A. McAllister.*

The *Leviathan*, pride of the American merchant marine and flagship of the United States Lines, Inc., New York, during a succession of heavy gales in the North Atlantic last fall, suffered serious damage to her shell plating and strength deck, as the result of heavy head seas and what was apparently an oversight in the original design of the ship as viewed with more recent knowledge on the subject. This damage was almost identical in extent with a similar misfortune which befell the White Star Liner *Majestic* described in the August, 1925, issue of "Marine Engineering and Shipping Age."

The *Leviathan* was withdrawn from active service for various repairs and improvements to both hull and machinery, including the removal and replacement of the damaged strength plating, the contract for which was awarded to the United Dry Docks, Inc., New York. This repair was unique because the location of the damage presented a problem never before encountered in the port of New York. It involved the driving of 3000 rivets ranging from 1¼ to 1½ inches in diameter on the side of the ship many feet above the water and in midwinter conditions of temperature. In order to drive these rivets under such difficulties, George H. Bates, vice-president of the United Dry Docks, Inc., invented what may be termed a strongback hydraulic bull riveter. It was first set up in the shops of the makers, the Staten Island plant of United Dry Docks, Inc., and specimen rivets driven simulating in every way the work to be done on the vessel.

The machine consists of two strongback members, each built up of double 1¼-inch thick plates, 8 feet 8 inches long by 6 feet 8 inches wide, welded together to form a 2½-inch face plate, which was reinforced by several 10-inch by ¾-inch vertical and horizontal stiffening webs welded to the back of the face plate and to each other. These strongbacks were so designed that one unit could be placed inside the ship's hull and the other outside and both held together by 12 chrome-nickel-steel tie rods each 1¼ inches in diameter run through frame rivet holes in the shell. Thus the strongbacks served as braces for the hydraulic heads which were similar to those used on the standard bull riveter and, being connected through the side shell, absorbed all of the pressure exerted by the hydraulic rams. The hydraulic heads were arranged to be portable and were suspended between the strongbacks on each side of the shell by means of block and tackle so that it was possible to drive approximately 200 rivets over a width of two frame spaces and nearly a full 'tween-deck height without changing the position of the strongbacks. Two complete sets of strongbacks were constructed, so that while one was in operation the second could be set up in a new position ready for driving. Each set weighed 8½ tons and required the erection of special means for handling from overhead on the vessel's topsides.

For operation of the hydraulic heads the usual auxiliary equipment such as accumulators, pumps, and piping was erected on the pier, and because of the cold weather during which the work was carried out, a special anti-freeze solution was required to maintain the equipment in working order. Special goose-neck dies were also made to facilitate the driving of rivets through frames and cramped quarters.

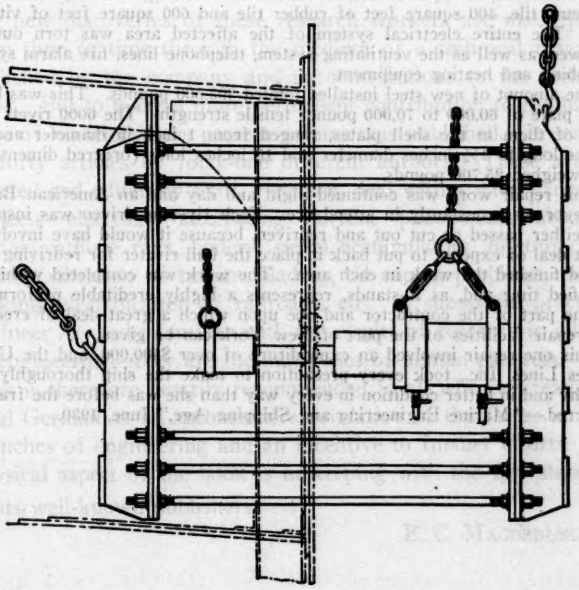
In all, six shell plates and forty-four deck plates were involved in the repair operation. Five of the shell plates about the No. 1 starboard expansion joint, in the main sheer strake and two strakes below were damaged, but it was necessary to remove a sixth plate because of the arrangement of butt-

* President, American Bureau of Shipping.

straps. These plates were 33 feet long by 78 inches wide and ranged in thickness from 1.10 inches to 1.36 inches. It was in these plates and their quadruple-riveted butt-straps that the strongback drove 3000 rivets, through varying thicknesses up to 4½ inches.

On (C) deck at the No. 1 expansion joint one stringer and doubler and one plate and doubler in (A) and (B) strakes were renewed. The damaged stringers were 33 feet long by 73 inches wide and 1⅛ inches thick, but in the new stringers the thickness was increased to 1½ inches. (A) and (B) strake plates and doublers were renewed as before, being 33 feet long by ¾-inch thick and ranging in width from about 61 to 80 inches.

To give the deck additional strength at this point two trebler plates were installed, one at the outboard after corner and one at the inboard after corner of the starboard uptake casing. At these corners the casing was cut away at the deck level to permit the trebler and strake plates to extend into the uptake opening and be tapered at an angle. The new treblers are 7 feet square and 1 inch thick. To increase the strength of the stringers and doublers, fifteen shell frames were cut at the (C) deck level, thus eliminating the necessity of notching these plates in way of the frames. To compensate for the cutting of the shell frames, a continuous gusset plate 30 inches wide by ⅝ inch thick was installed for a distance of 48 feet on the inboard toe of the frames and fastened to the frames with 6-inch by 6-inch by ½-inch angle clips and to the deck with a 6-inch by 6-inch by ⅝-inch angle for a distance of 48 feet.



STRONGBACK HYDRAULIC BULL RIVETER DESIGNED BY GEORGE H. BATES.

A new continuous 8-inch by 8-inch by ¾-inch stringer angle was installed for a distance of 57 feet and a new bounding-bar angle was installed along the after end of the uptake casing and continued forward on the outboard side for 15 feet. All new treble and quadruple riveted buttstraps ¾ inch thick were installed, one above each plate butt and one below each plate butt.

On the port side in the region of the No. 1 expansion joint similar repairs were effected on the (C) deck, including the stringer and doubler, which, while not damaged, were nevertheless renewed.

On (D) deck, starboard side, one cracked stringer and one cracked doubler plate were renewed. The stringer plate was 33 feet long by 70 inches wide and 1.04 inches thick and the doubler 33 feet by 64 inches by 0.82 inch. A new 6-inch by 6-inch by ⅝-inch waterway angle bar was installed at this point for a distance of 30 feet.

At the No. 2 expansion joint on (C) deck, starboard side, outboard of the uptake casing, several cracked plates and doublers were renewed. Other plates were renewed about the starboard elevator trunk where an additional trebler plate was installed. To further strengthen the deck in the region of the uptake casing and elevator trunk, a number of doublers and treblers were put down, and new 6-inch by 6-inch by ⅝-inch bounding angle bars were installed. Similar repairs were carried out on the port side of (C) deck.

The immensity of the reconditioning is emphasized by the fact that to get at the shell and deck steel work, it was necessary to dismantle 68 staterooms, 18 bathrooms, and 13 lavatories on (C), (D) and (E) decks, and remove and renew 10,000 square feet of asbestolith decking, 6000 square feet of linoleum tile, 400 square feet of rubber tile and 600 square feet of vitreous tile. The entire electrical system of the affected area was torn out and renewed, as well as the ventilating system, telephone lines, fire alarm system, plumbing and heating equipment.

The amount of new steel installed totaled 393,500 pounds. This was boiler shell plate of 60,000 to 70,000 pounds tensile strength. The 6000 rivets used, half of them in the shell plates, ranged from 1 inch in diameter and 2¼ inches long to 1½ inches diameter and 10 inches long (ordered dimensions) and weighed 25,700 pounds.

This repair work was continued night and day and an American Bureau surveyor was constantly in attendance. Each rivet as driven was inspected and either passed or cut out and redriven, because it would have involved a great deal of expense to put back in place the bull riveter for redriving after it had finished the work in each area. The work was completed within the specified time and, as it stands, represents a highly creditable performance on the part of the contractor and one upon which a great deal of credit to the repair facilities of the port of New York can be given.

This one repair involved an expenditure of over $400,000, and the United States Lines, Inc., took every precaution to make the ship thoroughly seaworthy and in better condition in every way than she was before the fracture occurred.—"Marine Engineering and Shipping Age," June, 1930.

BOOK REVIEW.

FORSHUNG UND TECHNIK (RESEARCH AND ENGINEERING), Edited by W. Petersen for the A. E. G. Co., Published by Julius Springer, Berlin—576 pages (8¼ by 11½ inches) 597 illustrations—price, bound R. M. 40.00.

One of the principal reasons for the success of the German A. E. G. Co. (Allgemeine Elekricitats-Gesellschaft) is its extensive research laboratories and the army of skilled engineers and scientists continuously engaged in forcing the secrets of nature and thus preparing the way for the industrial development of the new discoveries. Research is also responsible for the truly amazing variety of the activities of this vast industrial organization. The new discoveries of these research workers are made public from time to time through the medium of a technical magazine published by the company and as contributions to the technical press. But to make a more permanent contribution to the human store of knowledge the present volume was published. It consists of forty articles by forty-one different authors. These articles, selected and edited by Prof. Petersen, deal with a wide variety of subjects—electrons, alternating current phenomena, acoustics, talkies, cathode rays, boilers, steam accumulators, lightning protection, vibration of turbine blading, oil engine problems, electric cooking, electrification of railroads, etc. A successful practicing engineer has to be familiar with the results of research in all lines of engineering—they are the milestones of human progress. This book is therefore heartily recommended to all who can read technical German as an authoritative revelation of the status in many branches of engineering and an incentive to further efforts. The physical aspect of the book is in keeping with the fine standards of its well-known publishers.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

Rear Admiral Benjamin C. Bryan, U. S. Navy, died at the Naval Hospital, Washington, D. C., on July 21, 1930, at the age of 72 years. Admiral Bryan was one of the fifteen young Engineer Officers who met on October 15, 1888, and founded this Society. During his life he served at different times as Secretary-Treasurer, as a member of the Council and as President of the Society. His Naval career was most worthy and successful, including among other important assignments duty in the Bureau of Engineering, Engineer Officer at the Navy Yard, Philadelphia, Commandant at the Navy Yard, Charleston, and President of the Naval Examining and Retiring Board. It is with keen regret that the Society announces the passing of its illustrious member.

MEMBERSHIP.

The following members have joined the Society since the publication of the last previous number of the JOURNAL:

NAVAL.

Pederson, N. A., The La Bour Co., Inc., Elkhart, Ind.

CIVIL.

Aisenstein, Michael D., U. S. Iron Works, 2d and Jefferson Sts., Oakland, Calif.

Swain, Lloyd, 740 Front St., San Francisco, Calif.

Treiber, O. D., President, Treiber Diesel Engine Corporation, 815 Belmont Avenue, Collingswood, N. J.

ASSOCIATE.

Collom, S. H., Jr., Marchwood Apartments, Germantown, Philadelphia, Pa.

Morris, Charles, 163 Chestnut St., Rosebank, Staten Island, N. Y.

Winchester, Alexander, O'Brien Bros. Shipyard, Port Richmond, Staten Island, N. Y.

ERRATA.

Page 427, line 6—insert "axis of the" between "the" and "shaft".

Page 429, line 2—insert "to" between "proportional" and "the".

Page 429, line 17—change "connection" to "correction".

Page 429—line preceding Equation (4) should read "Taking moments about bearing B, *i. e.*, change "A" to "B".

Page 429—Equation (6) should read:

$$\text{"}A_A\ (1 + f) = R.\ a + A_B \times f \quad . \quad . \quad (6)\text{"}$$

Page 430—Equation (8) should read:

$$\text{"}A_A = \frac{R.\ a\ (e + l)}{L.\ l} = \frac{R.\ a}{L}\left(1 + \frac{e}{l}\right) \quad . \quad (8)\text{"}$$

Page 430—Equation (10) should read:

$$\text{"}B_B = \frac{R'.\ b}{L}\left(1 + \frac{f}{l}\right) \quad . \quad . \quad (10)\text{"}$$

[that part in parenthesis made no distinction between 1 and l and may have led to confusion in the use of the equation.]

Page 432, line 7—Insert "balance weight" between "auxiliary" and "would".

Page 441—Equation (8) should read:

$$\text{"}A_A = \frac{R.\ a}{L}\left(1 + \frac{e}{l}\right) \quad . \quad . \quad (8)\text{"}$$

Page 444—Equation (19) should read:

$$\text{"}A_A = R\left(1 + \frac{e}{l}\right) \quad . \quad . \quad . \quad (19)\text{"}$$

Page 444—Equation (21) should read:

$$\text{"}B_B = R'\left(1 + \frac{f}{l}\right) \quad . \quad . \quad . \quad (21)\text{"}$$

Page 448—Fig. 8, change "R" at top of figure to "R_A"
Fig. 10, change "Plane of R" to "Plane of R_B".

Page 449—In the space under "A–END", the 2nd line should read:

$$\text{"R.}\ \frac{a}{L} = 200 \times \frac{60}{65} = 185\ \text{"}\ \text{"} = A\text{"}$$

Page 449—In the space under "B–END", the 2nd line should read:

$$\text{"R'.}\ \frac{b}{L} = 300 \times \frac{55}{65} = 254\ \text{"}\ \text{"} = B\text{"}$$